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DESIGN AND TEST OF AN LHS LATERAL CONTROL SYSTEM FOR A T-2C AIR--ETC(U)
MAY 76 J N DEMARCHI, R K HANING N62269-75-C-0422
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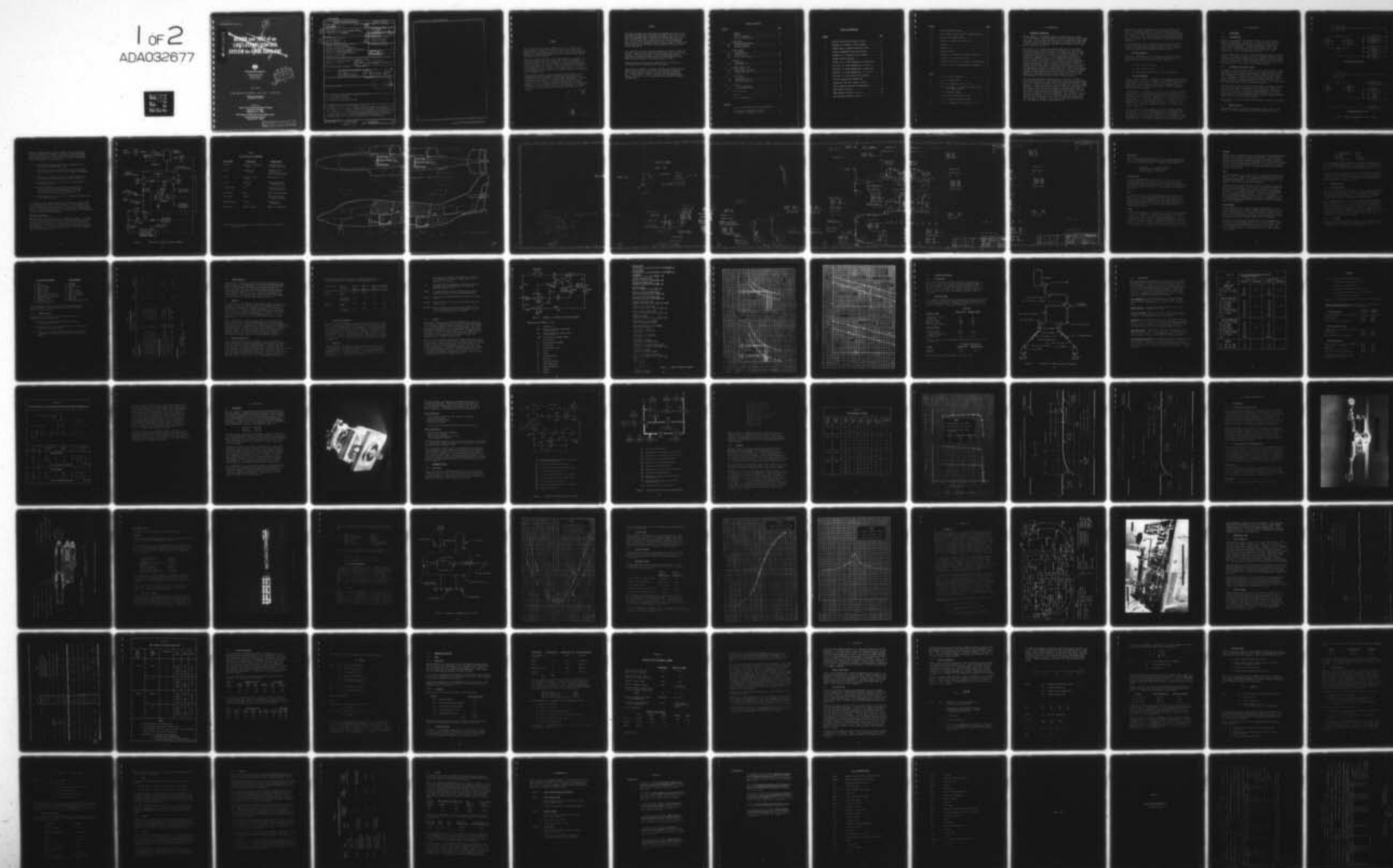

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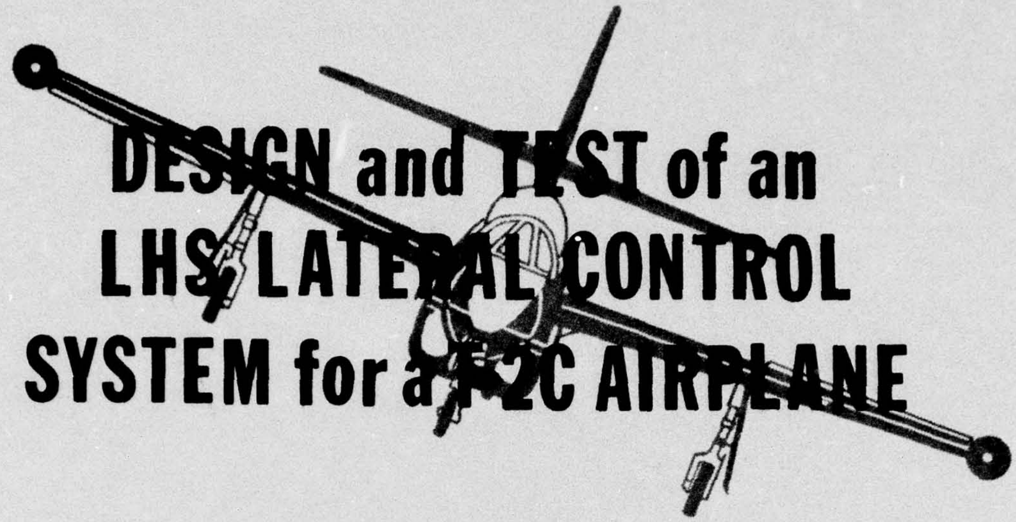

DESIGN and TEST of an LHS LATERAL CONTROL SYSTEM for a T-2C AIRPLANE

Rockwell International

Columbus Aircraft Division
4300 East Fifth Avenue
PO Box 1259
Columbus, Ohio 43216

MAY 1976

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Warminster, PA 18974

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

An 8000 psi hydraulic lateral control system was designed and tested for future evaluation in a T-2C airplane. Analyses were performed to estimate hydraulic fluid temperatures and pump suction line pressurization during flight. Laboratory tests were conducted to determine fluid temperatures, pressure surges, stability, and endurance. The tests demonstrated the system is satisfactory for progressing into the flight evaluation phase of the program.

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SUMMARY

This is Phase VIII of a program funded by the Navy to develop lightweight hydraulic systems for military aircraft. Prior phases have shown that use of an 8000 psi pressure level instead of the conventional 3000 psi level would significantly reduce weight and space requirements of hydraulic system components.

Endurance testing of 8000 psi aircraft type hardware was completed satisfactorily in Phase VII. The current phase involved preparations for flight evaluation of an 8000 psi hydraulic lateral control system in a Navy T-2C airplane. The test installation system was designed and flight test instrumentation were planned. The system contains the following major components rated for operation at 8000 psi: pump, relief valve, solenoid valve, aileron actuator, hoses, tubing, fittings, and MIL-H-83282 "less flammable" fluid.

Analyses were performed to estimate hydraulic fluid temperatures encountered during ground checkouts and inflight. Pump suction line pressurization was calculated for several operating conditions. System components were assembled in a laboratory setup simulating the T-2C test installation. System tests were performed to determine fluid temperatures, pressure surges, stability, and endurance. All operational characteristics of the laboratory setup were satisfactory. The system was therefore considered acceptable for flight evaluation in a T-2C airplane.

PREFACE

This report documents work performed by the Columbus Aircraft Division of Rockwell International Corporation in Columbus, Ohio 43216, for the Naval Air Development Center at Warminster, Pennsylvania 18974, under Contract N62269-75-C-0422. Technical direction was administered by Mr. J. Ohlson, Head, Fluid Systems Section, Air Vehicle Technology Department, Naval Air Development Center (30211), and Mr. N. Webb, Head, Fluid Systems Section, Mechanical Equipment Branch, Naval Air Systems Command (AIR-53031).

This report discusses the design and laboratory testing of an 8000 psi hydraulic lateral control system for future flight evaluation in a T-2C airplane. This work is related to tasks performed under Contract NOw-65-0567-d, N00019-68-C-0352, N00156-70-C-1152, N62269-71-C-0147, N62269-72-C-0381, N62269-73-C-0700, and N62269-74-C-0511.

Acknowledgement is given to Mr. B. Holland for his assistance on this project and his contributions to this report.

Appreciation is extended to the many individuals who provided helpful support and constructive criticism of the program; in particular, Mr. J. Crowder and Mr. N. Webb of the Naval Air Systems Command, Mr. J. Ohlson and Mr. J. Dever of the Naval Air Development Center, and Dr. F. Bellar, Jr. of the Columbus Aircraft Division of Rockwell International.

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1.0 INTRODUCTION

1.1 BACKGROUND INFORMATION

The development of a lightweight hydraulic system (LHS) for military aircraft has been a joint undertaking by the Navy and Rockwell International since 1966. The LHS concept involves the use of an 8000 psi pressure level to minimize the weight and space requirements of hydraulic components and lessen the severity of the weight and space restrictions present in high density, supersonic aircraft.

Prior phases of the program have examined many aspects of very high pressure hydraulic systems as applied to aircraft. The first phase was a theoretical study of pressure levels up to 20,000 psi, and concluded that operating pressures up to 9000 psi are feasible, Reference 1. The second phase consisted of: (1) a math model computer simulation to establish the dynamic response of two schematically simple hydraulic systems operating with pressures up to 12,000 psi; and (2) laboratory tests to confirm trends noted at lower pressures and gain operating experience with pressure levels up to 9000 psi, Reference 2. Phase III verified the math model dynamic response at 6000 and 9000 psi by means of laboratory tests conducted on a mass-loaded servo actuator powered by a very high pressure aircraft-type pump, Reference 3. The pump was designed and built by Abex Corporation in Oxnard, California, under the technical guidance of the Columbus Aircraft Division (CAD). The 9000 psi servo actuator was designed and fabricated by CAD and sized to simulate the RA-5C horizontal stabilizer flight control "muscle" actuator.

Phase IV involved hardware performance tests, selection of 8000 psi as the LHS operating pressure level, development of LHS design criteria, and use of these criteria in a study made to determine space and weight savings achieved if an 8000 psi hydraulic system were applied to the F-14 airplane, Reference 4. Phase V was an investigation of the detail performance characteristics of 8000 psi hardware including a variable delivery pump, 3 port solenoid valve, power servo actuator, and notched spool/sleeve type flow control valve operating with MIL-H-83282 fluid, Reference 5. In addition, the computer simulation of Reference 3 was updated and compared to hardware performance, and an industry-wide survey was made to locate 8000 psi static and dynamic seals. Phase VI consisted of preparations for conducting an endurance test on aircraft-type hardware designed for use in an 8000 psi hydraulic system, Reference 6.

Phase VII was a 100 hour endurance test conducted at 8000 psi and +200°F on lightweight hardware in a laboratory hydraulic system designed to be representative of aircraft-type circuitry. The hardware cycled were: pump, relief valve, restrictors, solenoid valves, flow control valve, seals (22), hydraulic fluid (MIL-H-83282), tubing, fittings, and hoses. The test was completed with no major problems, Reference 7.

The lightweight hydraulic system development program has shown that significant advantages can be gained, in terms of weight savings, reduced volume requirements, and lower overall costs, by operating at 8000 psi instead of the conventional 3000 psi level.

1.2 PROJECT OBJECTIVES

Objectives of Phase VIII of the LHS development program involved preparations for flight testing an 8000 psi lateral control system test installation on a T-2C airplane. Major tasks were:

(1) Test installation design

(2) Heat rejection analysis

(3) Laboratory compatibility test of system components

1.3 TECHNICAL APPROACH

The test system utilized an 8000 psi aileron power actuator designed and fabricated in Phase VII, Reference 7, and a GFE 8000 psi pump built under separate NADC contracts. Technical consultation required to support design and development of the pump was provided by CAD to the pump supplier, Abex. Other components used in the test system were procured in prior LHS phases.

The 8000 psi test installation for the T-2C lateral system was designed and preliminary layout drawings were made; flight test instrumentation requirements were planned. The system contains the following major components rated for operation at 8000 psi: pump, relief valve, solenoid valve, aileron actuator, hoses, tubing, fittings, and MIL-H-83282 "less flammable" fluid. A heat rejection analysis was performed to estimate hydraulic fluid temperatures and heat exchanger requirements. Calculations were completed to determine that pump suction line pressurization is adequate.

The test components were assembled in a laboratory system simulating the T-2C test installation. Operational tests were conducted on the pump and actuator. System tests were performed to determine fluid temperatures, pressure surges, and stability. System endurance testing was accomplished using a cycling schedule representative of the flight evaluation tests to be conducted in the next phase the LHS program.

2.0 SYSTEM DESIGN

2.1 DESCRIPTION

2.1.1 T-2C Airplane

The T-2C is a two-place, twin engine, subsonic jet trainer built by Rockwell International. The aircraft is designed for both land and carrier based operations and provides the means for a broad spectrum of military pilot training. Distinguishing features include wide track tricycle landing gear, straight tapered wings, and low slung intake ducts. The T-2C possesses excellent stability and control characteristics over a wide speed and "g" range.

The T-2C flight control system utilizes three primary surfaces: ailerons, elevators, and rudder. The ailerons are fully powered; elevator control is hydraulically boosted; rudder operation is manual. The flight control actuators are part of the mechanical linkage connecting the pilot's stick to the control surfaces. Thus, in the event of a hydraulic system malfunction, control of the aircraft can be achieved manually.

2.1.2 Existing T-2C Hydraulic System

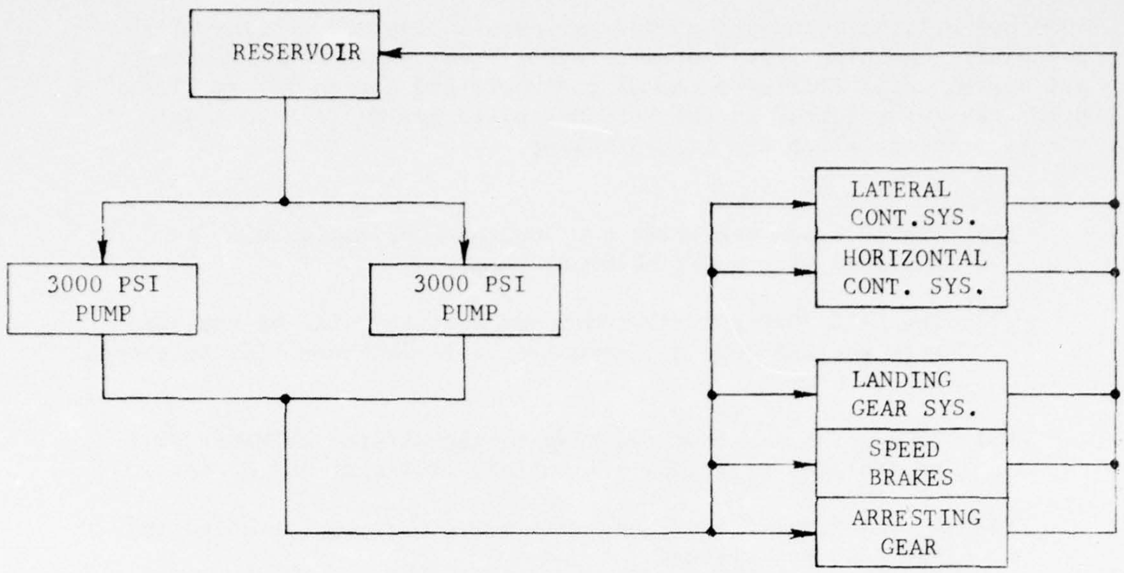
The T-2C has a 3000 psi, Type II (-65 to +275°F), single hydraulic system. Two pumps, one on each engine, provide power to operate the landing gear, speed brakes, arresting hook, aileron actuator, and elevator boost package. The pumps are constant pressure, variable delivery, axial piston units. Each pump is capable of delivering 4.5 gpm at 7800 rpm. Hydraulic fluid is supplied to the pumps by an air/oil type reservoir (1.14 gal. capacity) pressurized by engine bleed air. Two filters, one in the pressure and return lines, have five micron (absolute) ratings.

A priority valve is used to insure that the aileron and elevator actuators receive power if supply pressure drops below 1800 psi. A cockpit controlled shut-off valve is installed in the aileron/elevator sub-system to permit simulating loss of power for training purposes. The landing gear and arresting hook can be lowered and locked by gravity, if desired. The wheel brakes have an independent hydraulic system; a separate reservoir supplies fluid to the master brake cylinders.

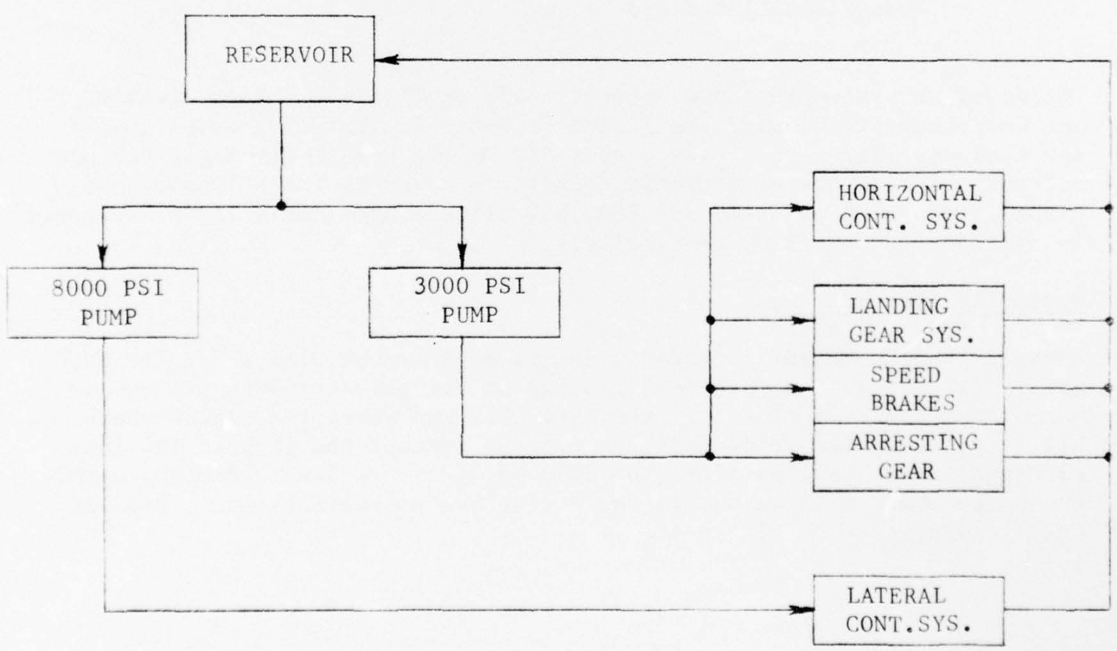
Block diagrams of the existing and modified hydraulic systems are presented on Figure 1.

2.1.3 Modified System

The modified system will operate at two pressure levels: 3000 psi and 8000 psi. An 8000 psi pump mounted on the L.H. engine will power an



EXISTING 3000 PSI SYSTEM



MODIFIED HYDRAULIC SYSTEM

FIGURE 1 EXISTING AND MODIFIED HYDRAULIC SYSTEMS

8000 psi aileron actuator; a 3000 psi pump on the R.H. engine will power all remaining T-2C hydraulic sub-systems. The 3000 and 8000 psi systems will utilize a common reservoir and common return lines. Major changes required in the T-2C hydraulic system to accommodate the test installation are listed below:

(1) The 3000 psi hydraulic pump on the L.H. engine will be replaced with a GFE 8000 psi pump.

(2) The T-2C 3000 psi aileron power actuator will be replaced with the LHS 8000 psi actuator built in Phase VII, Reference 7.

(3) 3000 psi lines from the pump to the aileron actuator will be replaced with 8000 psi tubing, fittings, and hoses.

(4) A relief valve and a shut-off valve will be installed in the 8000 psi system.

(5) A heat exchanger will be installed in the common case drain line of the two pumps because of (1) the small surface area and volume of the 8000 psi system and (2) the oversize 8000 psi pump, Section 3.1.

(6) MIL-H-5606 fluid in the T-2C will be replaced with MIL-H-83282 "less flammable" fluid.

The modified system is shown schematically on Figure 2. Specific 8000 psi components to be used are listed in Table I. Brief descriptions of the line routing, relief valve, shut-off valve, hoses, fittings, tubing, hydraulic fluid, and heat exchanger will be given in the following paragraphs. The 8000 psi pump and 8000 psi aileron actuator will be discussed in Sections 3.0 and 4.0, respectively.

Hydraulic Line Routing

General routing of the affected lines is depicted on Figure 3. The 8000 psi lines will follow identical routing of the original 3000 psi pressure tubing to a compartment aft of the fuel cell and above the engine where all affected system components are located (except the pump). The heat exchanger lines will parallel the 8000 psi pressure line. Routing within the compartment will be determined during system installation. Plumbing details are given on Figure 4.

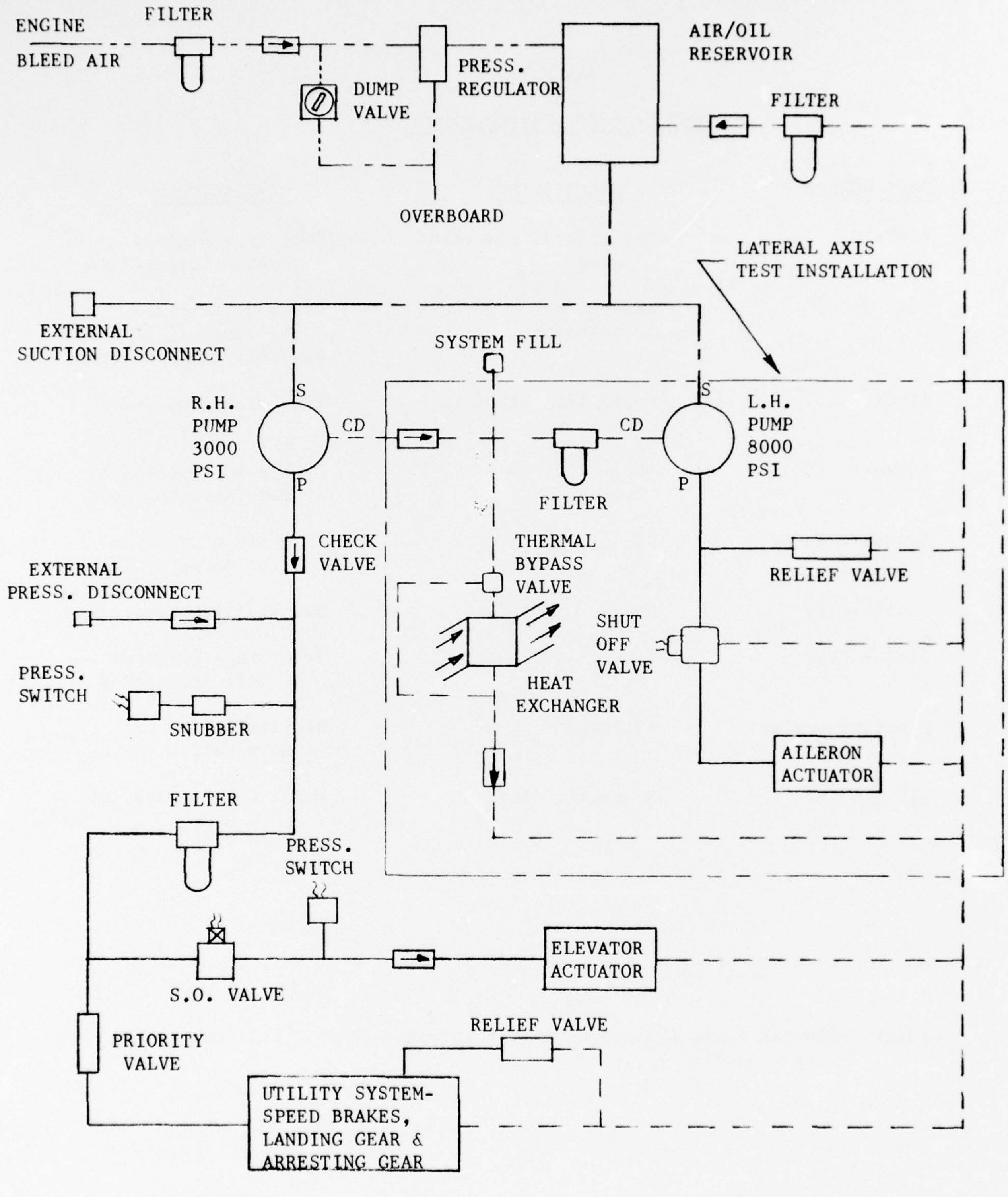


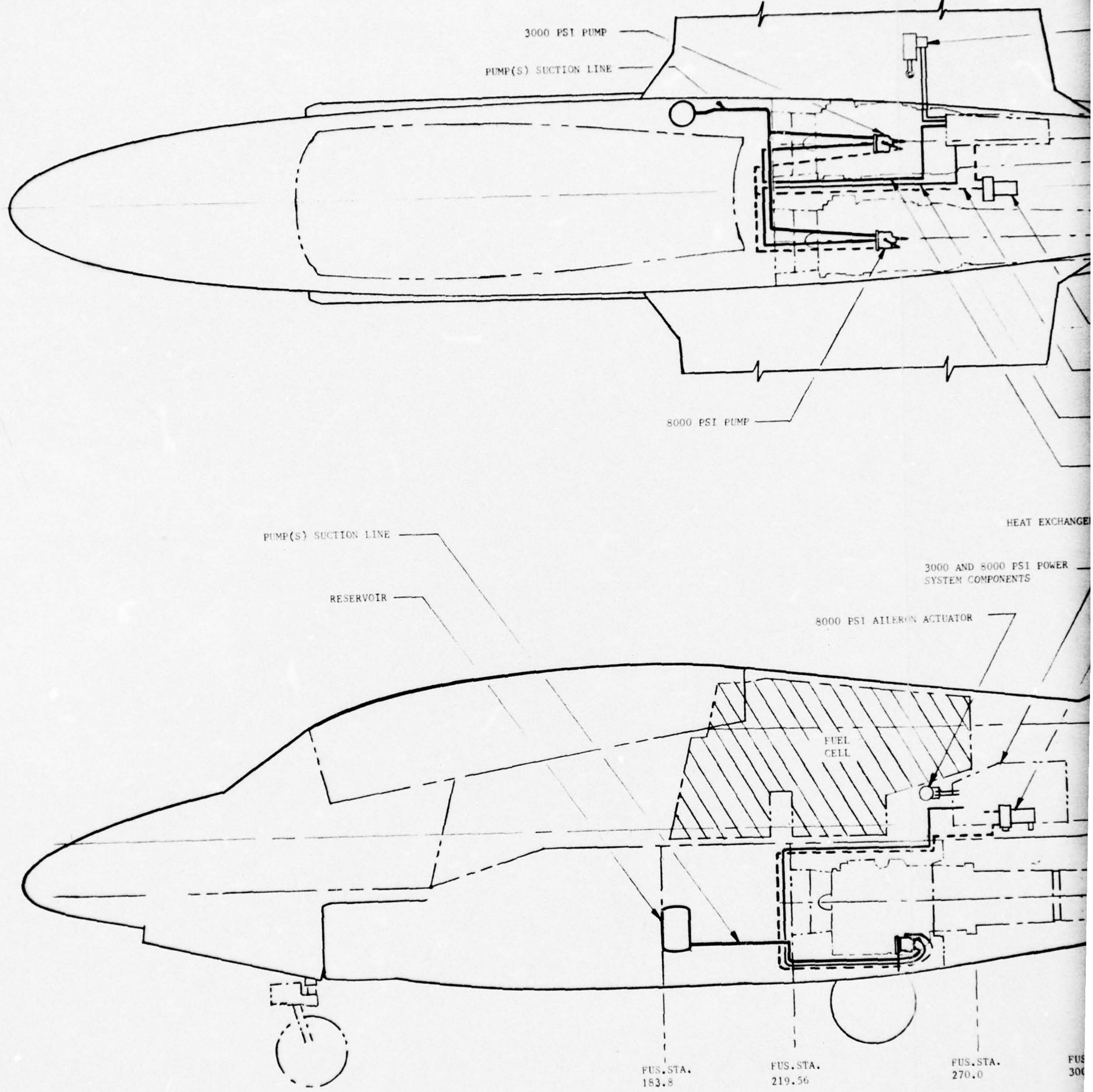
FIGURE 2 MODIFIED T-2C HYDRAULIC SYSTEM SCHEMATIC

TABLE I

LIST OF 8000 PSI COMPONENTS

PART NUMBER	DESCRIPTION	MANUFACTURER
APIV-106	Variable displacement pump	Aerospace Division of Abex Corporation
4257-01	Aileron power actuator	Columbus Aircraft Division of Rockwell International
1180A	Hydraulic relief valve	Pneu Draulics, Inc.
15390-1	Shut-Off valve	Sterer Engineering & Manufacturing Co.
37404004-0264C	Hose	Titeflex Division of Atlas Corporation
R44598-0310	Hose	Resistoflex Corporation
21-6-9 CRES	Tubing	Trent Tube Division of Colt Industries
Dynatube Series	Fittings	Resistoflex Corporation
MIL-H-83282	Hydraulic Fluid	Mobil Oil Corporation

NOTE: 8000 psi instrumentation is not included in this listing, reference Section 2.2.



3000 PSI PUMP
PUMP(S) SUCTION LINE
8000 PSI PUMP
HEAT EXCHANGE
3000 AND 8000 PSI POWER SYSTEM COMPONENTS
8000 PSI AILERON ACTUATOR
PUMP(S) SUCTION LINE
RESERVOIR
FUEL CELL
FUS.STA. 183.8
FUS.STA. 219.56
FUS.STA. 270.0

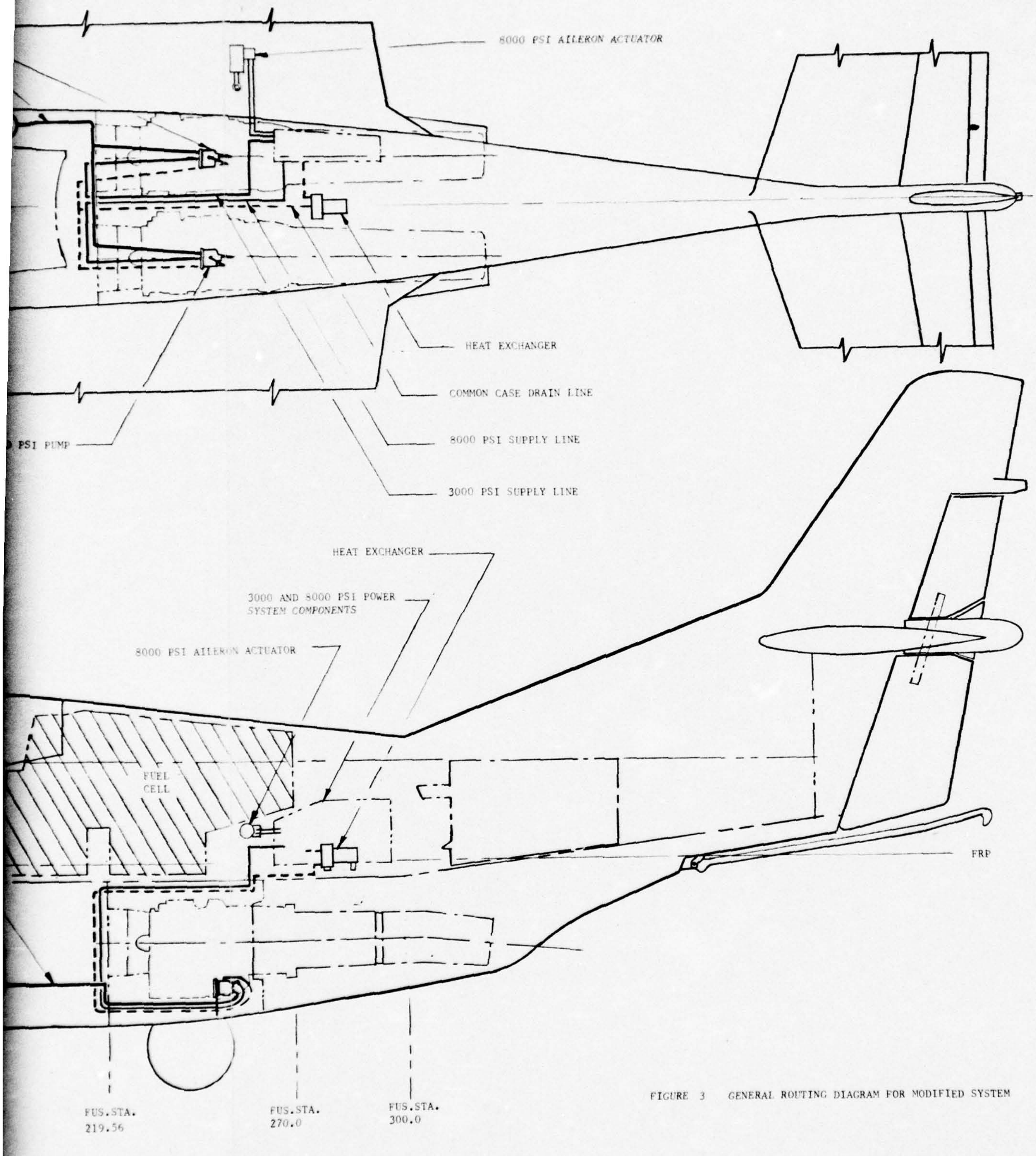
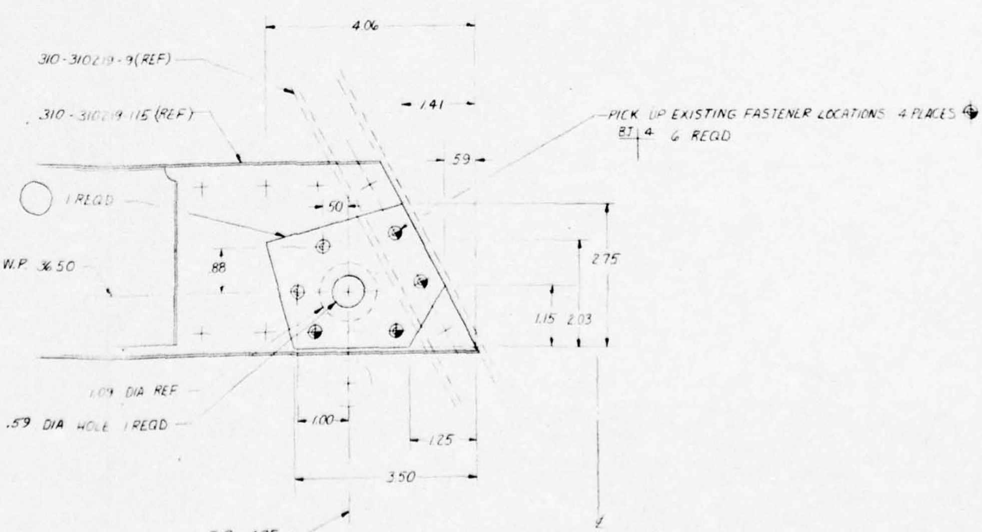


FIGURE 3 GENERAL ROUTING DIAGRAM FOR MODIFIED SYSTEM

24 23 22 21 20



VIEW A-A

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(FULL SIZE)

R44359P-06 1 REQ
R44118T-06 1 REQ
LD153-0011-0024 1 REQ

24 23 22 21 20

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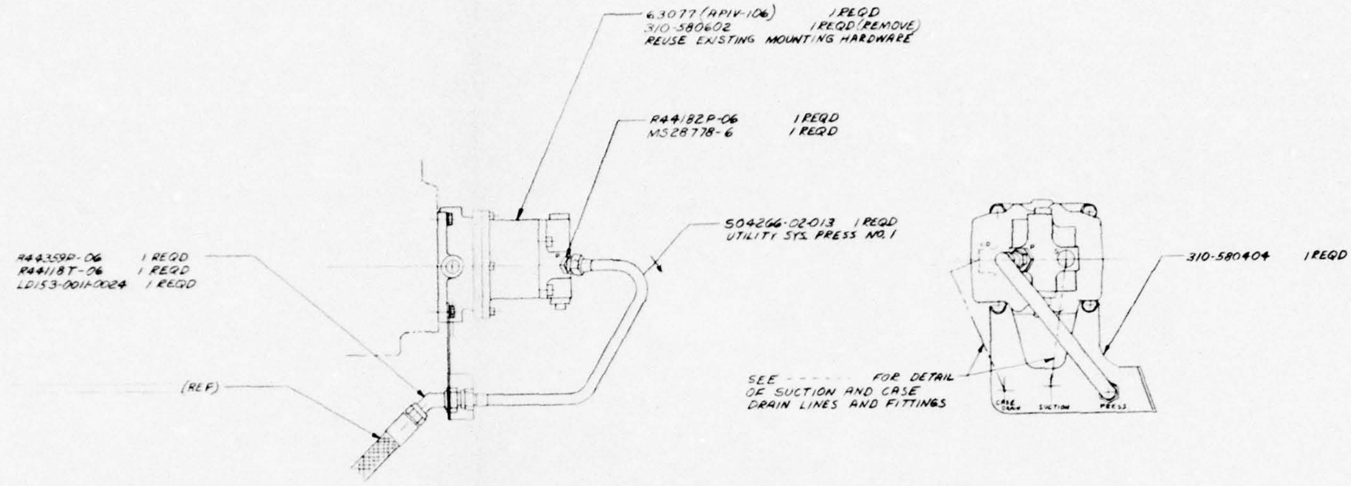

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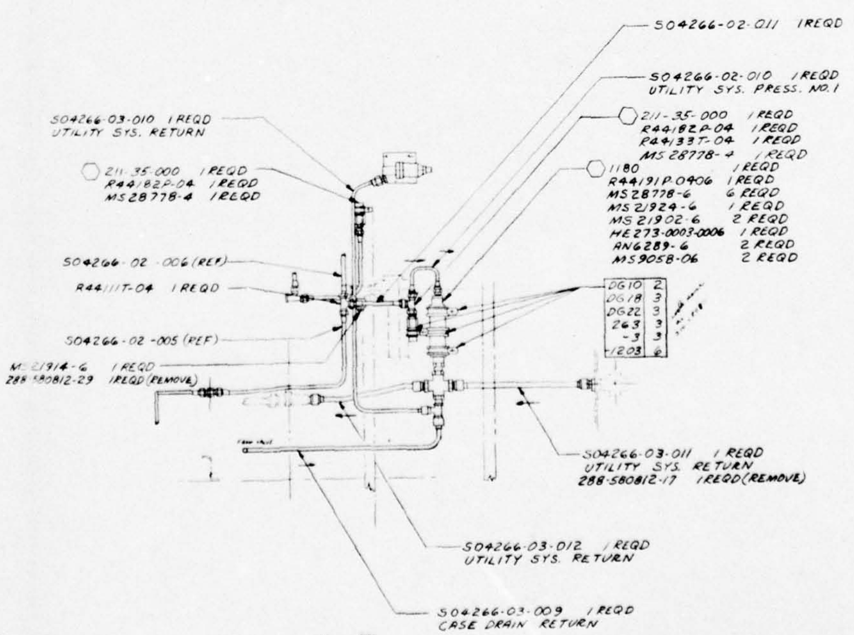

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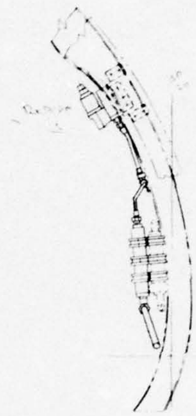
VIEW 13 - 13

2

20	19	18	17	16	15

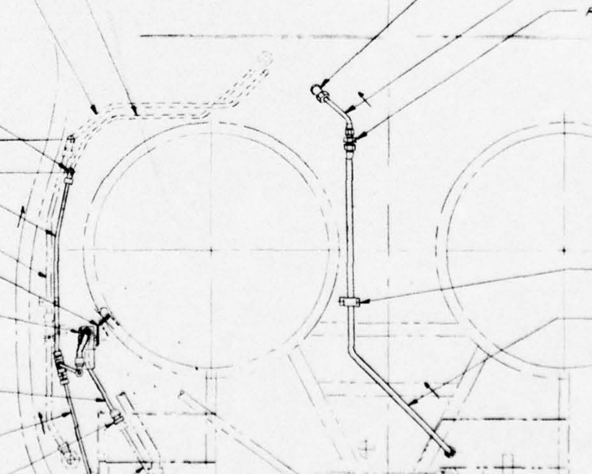
15	14	13	12	11



310-580404 1 REQD

-011 1 REQD

010 1 REQD
PRESS. NO. 1



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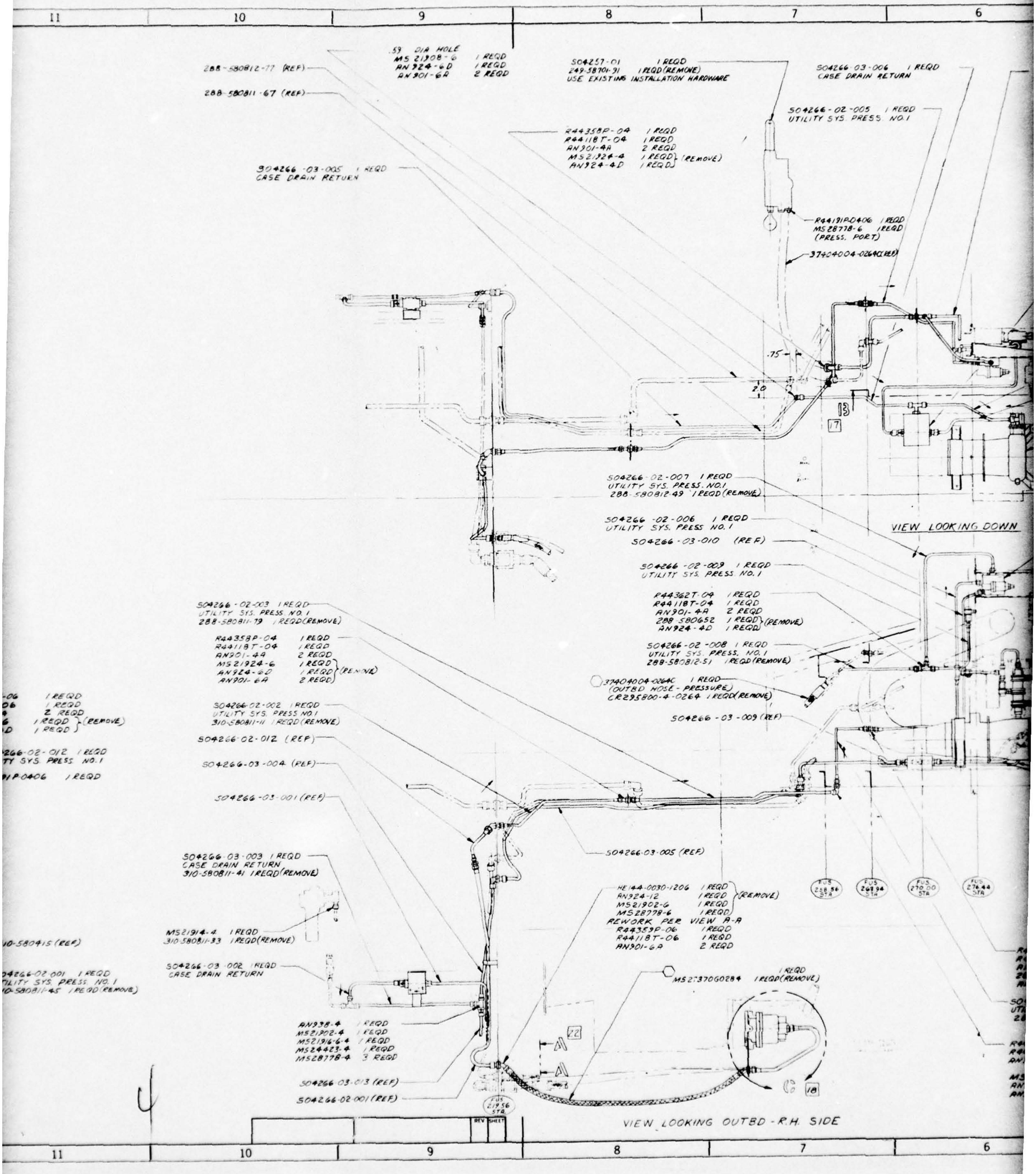

11
10
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8
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6
.59 DIA HOLE
MS 21908-6 1 REQD
AN924-6D 1 REQD
AN901-6A 2 REQD
288-580812-77 (REF)
288-580811-67 (REF)
S04257-01 1 REQD
249-58701-91 1 REQD (REMOVE)
USE EXISTING INSTALLATION HARDWARE
S04266-03-006 1 REQD
CASE DRAIN RETURN
S04266-02-005 1 REQD
UTILITY SYS PRESS NO. 1
R44358P-04 1 REQD
R44118T-04 1 REQD
AN901-4A 2 REQD
MS21924-4 1 REQD (REMOVE)
AN924-4D 1 REQD
S04266-03-005 1 REQD
CASE DRAIN RETURN
R44191P-0406 1 REQD
MS28778-6 1 REQD
(PRESS. PORT)
37404004-0264C (REF)
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2.0
13
17
S04266-02-007 1 REQD
UTILITY SYS. PRESS. NO. 1
288-580812-49 1 REQD (REMOVE)
S04266-02-006 1 REQD
UTILITY SYS. PRESS NO. 1
S04266-03-010 (REF)
VIEW LOOKING DOWN
S04266-02-009 1 REQD
UTILITY SYS. PRESS NO. 1
R44362T-04 1 REQD
R44118T-04 1 REQD
AN901-4A 2 REQD
288-580652 1 REQD (REMOVE)
AN924-4D 1 REQD
S04266-02-008 1 REQD
UTILITY SYS. PRESS. NO. 1
288-580812-51 1 REQD (REMOVE)
37404004-0264C 1 REQD
(OUTBD HOSE-PRESSURE)
CR295800-4-0264 1 REQD (REMOVE)
S04266-03-009 (REF)
S04266-02-003 1 REQD
UTILITY SYS. PRESS. NO. 1
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R44358P-04 1 REQD
R44118T-04 1 REQD
AN901-4A 2 REQD
MS21924-6 1 REQD
AN924-6D 1 REQD (REMOVE)
AN901-6A 2 REQD
S04266-02-002 1 REQD
UTILITY SYS. PRESS NO. 1
310-580811-11 1 REQD (REMOVE)
S04266-02-012 (REF)
S04266-03-004 (REF)
S04266-03-001 (REF)
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6 1 REQD (REMOVE)
D 1 REQD
266-02-012 1 REQD
TY SYS. PRESS. NO. 1
91P-0406 1 REQD
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S04266-03-003 1 REQD
CASE DRAIN RETURN
310-580811-41 1 REQD (REMOVE)
FUS 258.56 STA
FUS 268.94 STA
FUS 270.00 STA
FUS 276.44 STA
HE144-0030-1206 1 REQD
AN924-12 1 REQD (REMOVE)
MS21902-6 1 REQD
MS28778-6 1 REQD
REWORK PER VIEW A-A
R44358P-06 1 REQD
R44118T-06 1 REQD
AN901-6A 2 REQD
MS21914-4 1 REQD
310-580811-33 1 REQD (REMOVE)
10-580415 (REF)
S04266-03-002 1 REQD
CASE DRAIN RETURN
04266-02-001 1 REQD
TILITY SYS. PRESS. NO. 1
10-580811-45 1 REQD (REMOVE)
MS27370G0284 1 REQD (REMOVE)
AN938-4 1 REQD
MS21902-4 1 REQD
MS21916-6-4 1 REQD
MS24423-4 1 REQD
MS28778-4 3 REQD
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18
S04266-03-013 (REF)
S04266-02-001 (REF)
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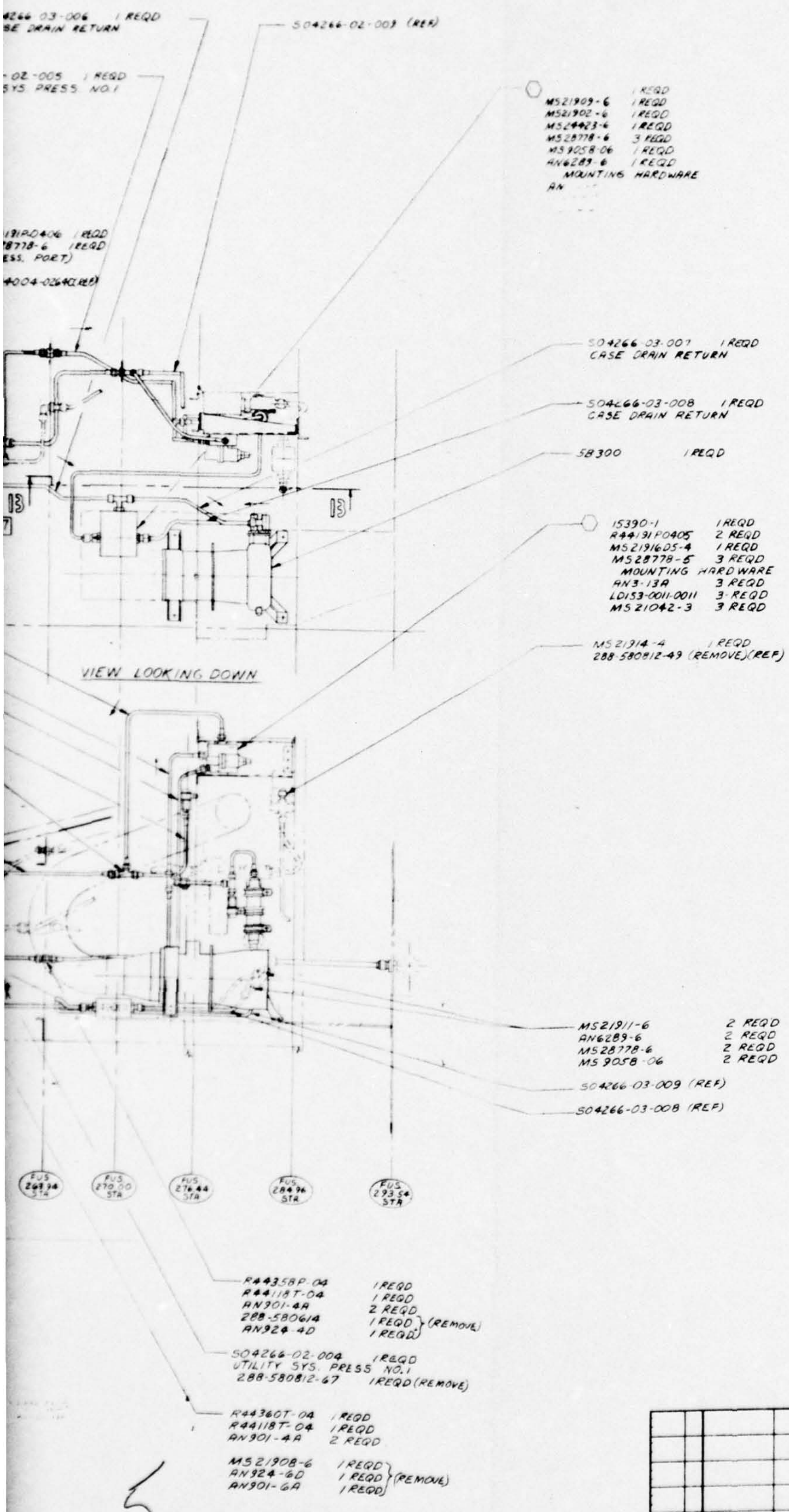

FIGURE 4

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.041 THRU .130	+.002 -.001
.131 THRU .229	+.003 -.001
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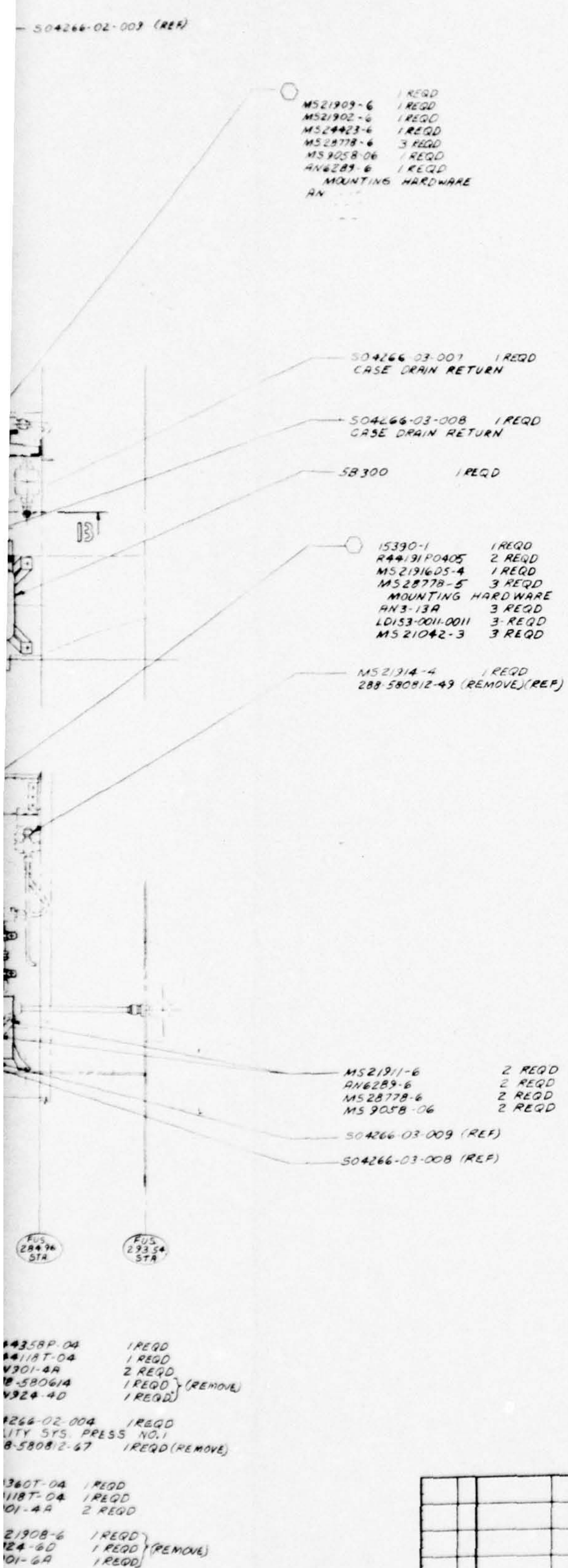

FIGURE 4

REVISIONS

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.041 THRU .130	+.002 −.001
.131 THRU .229	+.003 −.001
.230 THRU .500	+.004 −.001
.501 THRU .750	+.005 −.001
.751 THRU 1.000	+.007 −.001
1.001 THRU 2.000	+.010 −.001
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Columbus Aircraft Division
Columbus, Ohio 43216

HYDRAULIC EQUIP INSTALLATION 8000PSI SYSTEM FOR T-2C AIRCRAFT

SIZE	CODE IDENT NO.	
J	89372	S04266-01

SCALE 1/4 [illegible] SHEET

ITEM	QTY	NEXT ASSY	USED ON	END ITEM NO.	THRU	HEAT TREAT
REQD PER END ITEM		APPLICATION		EFFECTIVITY		FINISH

6

Relief Valve

Relief valve P/N 1180A was procured for use in a prior test program where the unit functioned satisfactorily for 100 hours operating at 8000 psi and +200°F, Reference 6. Performance requirements are:

Rated Flow:	5 gpm at 8800 psi
Cracking Pressure:	8600 ± 100 psi
Proof Pressure:	12,000 psi

Shut-Off Valve

Solenoid valve P/N 15390-1 was procured for use in a prior test program, Reference 5. The valve is a normally closed, 3 port, 2 position, pilot operated unit powered with 28 VDC. The valve will permit depressurizing the aileron actuator, and be cockpit controlled. This unit performed satisfactorily in the endurance test reported in Reference 7.

Hoses

The 3000 psi hoses used (1) near the pump and (2) at the aileron actuator will be replaced with similar sized 8000 psi hoses. The pump hose will be Resistoflex R44598 series; the actuator hose will be Titeflex 370 series. Both hoses will have end fittings which mate with Resistoflex "Dynatube" connectors. Both hose types have been tested at 8000 psi by the Naval Air Development Center at Warminster, Pennsylvania.

Tubing

Tubing used in the 8000 psi circuit will be identical to that developed for the LHS endurance test reported in Reference 7. The tubing is 1/8 hard, 21 Cr - 6 Ni- 9 Mn stainless steel designed for a burst pressure of 3 X 8000 = 24000 psi. Room temperature tensile strength is 142,000 psi; yield strength is 120,000 psi; and fatigue strength is 49,000 psi. Tubing sizes used in 8000 psi circuits will be 1/4 X .025 and 3/8 X .038. Lines used in low pressure circuits will be standard 6061-T6 aluminum alloy tubing; sizes will be 1/4 X .035 and 3/8 X .049.

Fittings

Fittings used in the 8000 psi circuit will be Resistoflex "Dynatube" series similar to those evaluated in Phase III, Reference 7. 6AL-4V titanium fittings will be mechanically swaged to each line assembly. Titanium fittings swaged on 21-6-9 tubing successfully passed flexure testing at 8000 psi conducted by Resistoflex, Appendix A. All fittings in low pressure circuits will be standard MS flareless types; in general, steel fittings will be used where steel tubing is required, and aluminum fittings where aluminum lines are used.

Fluid

MIL-H-83282 hydraulic fluid will be used in the LHS flight demonstration program. This is a shear stable, synthetic hydrocarbon rated for use at temperatures up to +400°F. It is compatible with MIL-H-5606, but less flammable than MIL-H-5606. MIL-H-83282 was used successfully in a 100 endurance test at 8000 psi and +200°F, Reference 7.

MIL-H-5606 will be removed from the existing T-2C hydraulic system and replaced with MIL-H-83282. The system will be drained as thoroughly as possible, however no attempt will be made to eliminate all traces of MIL-H-5606. It is estimated that approximately 95% of the MIL-H-5606 fluid can be removed without "purging". A 5% dilution of MIL-H-83282 is not considered detrimental to either its chemical or physical properties with the exception of fire resistance. The presence of more than approximately 8% of MIL-H-5606 will lower the flammability characteristics of MIL-H-83282 to that of MIL-H-5606, which in this application is not significant.

Heat Exchanger

A heat exchanger will be required to keep fluid temperatures in the "normal" operating range of +160 to +200°F, reference Section 2.3. The oversized pump (Section 3.1) combined with the small surface area and volume of the 8000 psi system result in the need for cooling. A heat removal rate of approximately 122 BTU/min should be adequate for ground operations with less heat removal required at higher altitudes.

An oil-to-air heat exchanger compatible with cooling requirements was located in the military supply system. This unit, P/N 5B300, is manufactured by Standard Thompson and has an electric motor driven fan. It is recommended that this assembly be made available as GFE for the flight evaluation program. Performance details of the heat exchanger are as follows:

Oil temperature in:	+220°F
Air temperature in:	+90°F
Oil flow:	1 gpm
Air flow:	16 lb/min
Heat removal rate:	148 BTU/min

The heat exchanger will be installed in the common case drain line of the 3000 and 8000 psi pumps, Figure 2, and located in the compartment aft of the fuel cell above the engines, Figure 3. Outside air will be ducted to the heat exchanger inlet from an opening in the side of the aircraft.

Depending on operating conditions, the heat exchanger could (without control) remove too much heat from the system. To prevent overcooling, a bypass valve will be installed between the inlet and outlet ports. This will automatically bypass fluid around the heat exchanger when fluid temperatures are below +130 ± 10°F.

2.2 INSTRUMENTATION

2.2.1 Recording Systems

T-2C BuNo. 152383 is retained at CAD as a bailed flight test vehicle. This aircraft has been used in numerous special evaluation programs and is equipped with flight data acquisition systems. Two systems will be used in the LHS program: (1) a 50 channel oscillographic recorder system; and (2) a 21 hole photo recorder system.

Pilot instrumentation controls are located on the cockpit instrument panel and on the control stick. Oscillographic paper speed and/or photo recorder frame rate can be selected by switches on the instrument panel. A push button switch on the pilot's control stick turns on all recording devices programmed to operate. A second switch on the stick causes an oscillographic trace to deflect and turns on an event light in the photo recorder. This provides a means of correlating data between recording systems. All oscillographic paper and photo recorder film will be processed at CAD.

2.2.2 Parameters

Existing instrumentation includes parameters not applicable to the LHS program; this equipment will not be removed. Currently installed instrumentation will record the following parameters:

OSCILLOGRAPH RECORDER	PHOTO RECORDER
1. OAT	1. Airspeed
2. Correlation	2. Altitude
3. Pilot's marker	3. Pilot's marker
4. Angle of pitch	4. Time
5. Angle of bank	5. Frame counter
6. Rate of roll	6. Angle of attack
7. Normal acceleration at CG	7. Angle of sideslip
8. Longitudinal stick position	8. Fuel used, L/R
9. Lateral stick position	9. EGT, L/R
10. Rudder position	10. RPM, L/R
11. L/R aileron position	11. Fuel temperature, L/R
12. Elevator position	

New instrumentation added specifically for the LHS program are listed on Table II. Operating range, accuracies, and response capabilities are also listed.

2.2.3 Cockpit Controls

Equipment will be installed to permit the pilot to monitor the status of the 8000 psi system.

(1) A switch will be provided to operate the aileron actuator shutoff valve.

(2) An indicator will provide direct readout of system pressure upstream of the shutoff valve.

(3) A "hot" caution light will operate when hydraulic fluid temperature in the pump suction line exceeds approximately +200°F.

TABLE II

LIST OF INSTRUMENTATION

PARAMETER	TRANSDUCER	RANGE	ACCURACY	RESPONSE
PHOTO RECORDER				
1. Temp, fuel access bay	TC	-20 to +160°F	±3°	2 Hz
2. Temp, LH pump suction port	TC	0 to +240°F		
3. Temp, LH pump case drain port	TC	+60 to +275°F		
4. Temp, RH pump case drain port	TC	+60 to 275°F		
5. Temp, heat exchanger inlet	TC	+60 to +275°F		
6. Temp, heat exchanger outlet	TC	+60 to +275°F		
OSCILLOGRAPH RECORDER				
1. Press, LH pump suction port	SGPT	0 to 50 PSIA	±1%	60 Hz
2. Press, LH pump case drain port	SGPT	0 to 100 PSIA		
3. Press, Aileron actuator inlet line	SGPT	0 to 10,000 PSI		
4. Flow, LH pump case drain line	TFM	0 to 1 GPM	±1%	2 Hz

NOMENCLATURE:

TC Thermocouple
SGPT Strain gage pressure transducer
TFM Turbine flow meter

2.3 THERMAL ANALYSIS

The existing T-2C 3000 psi single hydraulic system is powered by two parallel pumps. Fluid temperatures seldom exceed +220°F during normal operation with maximum temperatures occurring in the pump case drain lines. The modification to incorporate the 8000 psi system will essentially split the pressure distribution lines into two subsystems within the single hydraulic system -- the 8000 psi loop being much smaller than the 3000 psi loop. The relatively small surface area of the 8000 psi system plus the heat rejection of the oversized 8000 psi pump (Section 3.1) made a thermal analysis necessary.

2.3.1 Approach

All input power to the hydraulic system pumps is either utilized as useful work or rejected in the form of heat. The generated heat must be dissipated by transfer to the environment (air or structure) and/or removed by a heat exchanger. When the generation rate (steady state) equals the heat loss rate, the system has reached thermal equilibrium. Time to reach the equilibrium temperature is a function of the generation rate, heat loss rate, fluid and component mass, etc. For the purposes of this analysis, it is assumed that the system has reached equilibrium.

Many of the hydraulic system components involved in the steady state mode of operation are located in compartments which are above ambient temperature in flight. Compartment air temperatures reach approximately +120°F(max.)during ground operation where steady state flow is somewhat less than airborne flow. (Steady state flow consists primarily of system internal leakage.) The 8000 psi hardware are located in the same compartments, and the lines follow the same routing as the 3000 psi components which were removed, therefore the modified system will experience the same environment as the original 3000 psi system.

2.3.2 Operating Conditions

Various operating conditions were examined in order to establish the overall heat transfer coefficient required in the analysis. All temperatures were based on an ICAO standard atmosphere. Since the analysis assumes steady state operating conditions, all flow (power) delivered by the two pumps is assumed converted to heat. In order to check the validity of the analysis, calculations were made of the original T-2C 3000 psi system for comparison with existing flight test data. The majority of the temperature data available were at 35,000 feet, therefore calculations were made for this altitude. The LHS flight demonstration will be

conducted below 20,000 feet, however, reference Section 2.4.3. Operating conditions used in the analysis are summarized below:

Configuration	Flight Condition	Outside Air Temp., °F	Comp'm't Air Temp., °F	System Demand (gpm)	
				3000 psi	8000 psi
Original T-2C	Ground Idle	+70°	+120°	.1	--
"	Stabilized @ 35,000 ft	-65°	0°	.2	--
Modified T-2C	Ground Idle	+70°	+100°	.07	.09
"	Stabilized @ 15,000 ft	+23°	+50°	.3	.09

2.3.3 Heat Balance Equations

The system used for analysis is depicted schematically on Figure 5. The need for cooling was anticipated, therefore a heat exchanger was placed in the common case drain line of the two pumps. Using the temperatures, flows, and surface areas designated on Figure 5, heat balance equations for each line and junction were written. Equations for pump discharge fluid temperature as a function of inlet temperature were written based on laboratory test data. The thermal analysis equations are listed on Figure 6. The equations were re-written for solution using determinants and a digital computer program was developed. The program is presented in Appendix B.

2.3.4 Input Data

Component surface areas were calculated using detail drawings and preliminary layouts of the 8000 psi circuits; these areas were written into the program as constant values. The specific heat and density of MIL-H-83282 fluid were assumed constant -- 0.55 BTU/lb/°F and 0.0297 lb/in^3, respectively. Other parameters were also treated as constants but were changed from case to case as follows:

h	The overall heat transfer coefficient was assumed constant over the entire system. Values from 1 to 8 $BTU/hr/ft^2/°F$ were used.
T_A	Air temperature was assumed to be the same in all compartments, and varied according to the operating condition, Section 2.3.2.
BTUHE	Cooling provided by the heat exchanger was varied to determine the effect on system fluid temperatures.
QC3,QC8	Pump case drain flows were based on test data, Section 3.2.2.
QD3,QD8	Pump discharge flows varied with operating conditions, Section 2.3.2.
BTU3,BTU8	Pump heat rejection was based on test data, Section 3.2.2. Estimates were made for those operating conditions where test data were not available.

2.3.5 Results

The results of five runs are tabulated in Appendix B. Runs 1 and 2 were made to estimate effective overall heat transfer coefficients of the system by comparing computed temperatures with actual temperatures measured during past T-2C flight test programs. The overall heat transfer coefficients were thus determined to be approximately 6 $BTU/hr/ft^2/°F$ during ground idle and 5 to 5.5 in flight at 35,000 ft., Figures 7 and 8. The ground idle coefficient may be "high" if temperature stabilization was not achieved when the data were recorded (after a 10 minute warmup period).

Runs 3, 4 and 5 were made using the heat transfer coefficients derived in runs 1 and 2. Figures 9 and 10 show calculated fluid temperatures in the reservoir and case drain line of the 8000 psi pump as a function of BTU's removed by the heat exchanger. Using a coefficient of 6 during ground idle, then 7300 BTU/hr (122 BTU/min) must be removed by the heat exchanger to maintain a stabilized temperature of +180°F in the reservoir, Figure 9. Using a coefficient of 5 for flights at 15,000 ft, approximately 4500 BTU/hr (75 BTU/min) must be removed to maintain the reservoir at +180°F, Figure 10.

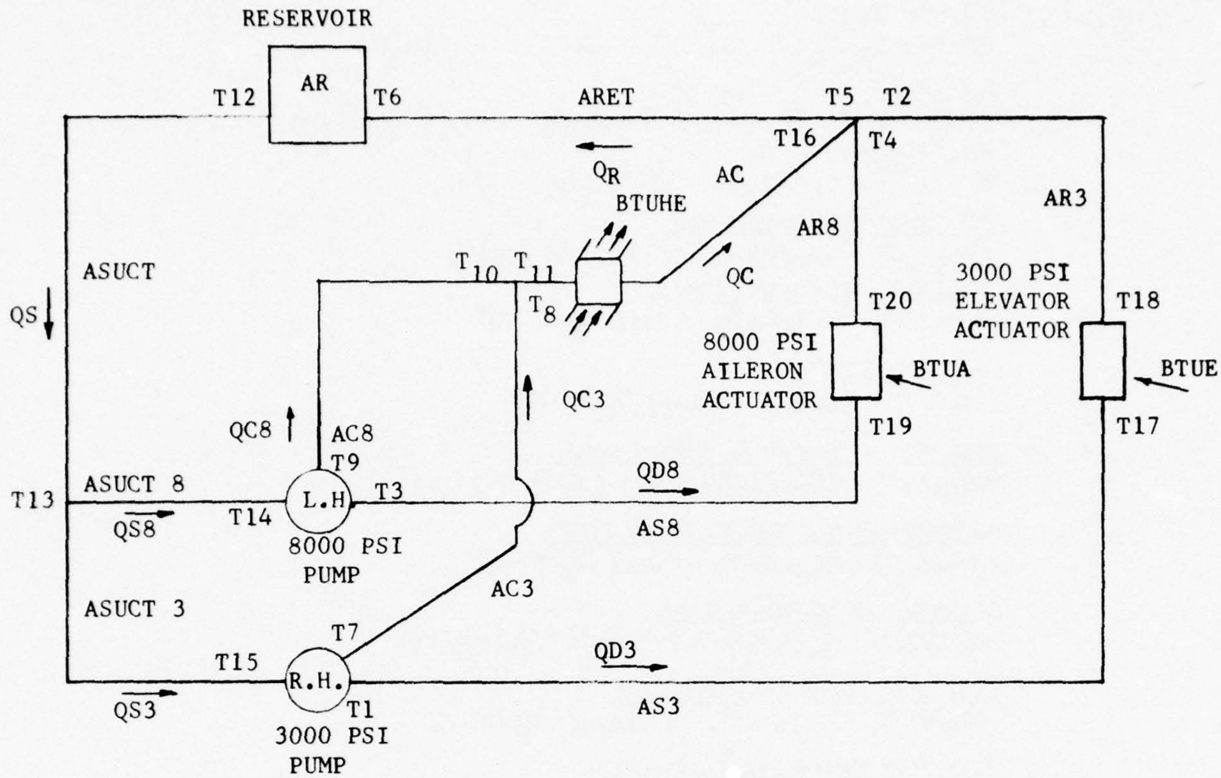


FIGURE 5 MODIFIED T-2C SYSTEM HEAT BALANCE DIAGRAM

NOMENCLATURE FOR FIGURES 5 AND 6

A	Surface area, ft^2
BTUA	Heat rejected by aileron actuator, BTU/hr (work out = zero)
BTUE	Heat rejected by elevator actuator, BTU/hr (work out = zero)
BTUHE	Heat removed by heat exchanger, BTU/hr
C	Case drain line
C_p	Fluid specific heat, BTU/lb/°F
D	Discharge line
K	Conversion factor = 411.5
P_3	3000 psi
P_8	8000 psi
Q	Fluid flow rate, gpm
R	Return line
S	Suction line or system
T	Fluid temperature, °F
T_A	Ambient temperature, °F
3	3000 psi
8	8000 psi

8000 PSI PUMP

$$BTU8 + KQ_{S8} C_p T_{14} = KQ_{C8} C_p T_9 + KQ_{D8} C_p T_3 + hA_{P8}\left(\frac{T_3 + T_9 + T_{14}}{3} - T_A\right)$$

3000 PSI PUMP

$$BTU3 + KQ_{S3} C_p T_{15} = KQ_{C3} C_p T_7 + KQ_{D3} C_p T_1 + hA_{P3}\left(\frac{T_1 + T_7 + T_{15}}{3} - T_A\right)$$

RESERVOIR

$$KQ_R C_p T_6 = KQ_S C_p T_{12} + hA_R\left(\frac{T_6 + T_{12}}{2} - T_A\right)$$

3000 PSI PRESSURE LINES

$$KQ_{D3} C_p T_1 = KQ_{D3} C_p T_{17} + hA_{S3}\left(\frac{T_1 + T_{17}}{2} - T_A\right)$$

8000 PSI PRESSURE LINES

$$KQ_{D8} C_p T_3 = KQ_{D8} C_p T_{19} + hA_{S8}\left(\frac{T_3 + T_{19}}{2} - T_A\right)$$

COMMON RETURN

$$KQ_R C_p T_5 = KQ_R C_p T_6 + hA_{RET}\left(\frac{T_5 + T_6}{2} - T_A\right)$$

3000 PSI PUMP CASE DRAIN LINE

$$KQ_{C3} C_p T_7 = KQ_{C3} C_p T_8 + hA_{C3}\left(\frac{T_7 + T_8}{2} - T_A\right)$$

8000 PSI PUMP CASE DRAIN LINE

$$KQ_{C8} C_p T_9 = KQ_{C8} C_p T_{10} + hA_{C8}\left(\frac{T_9 - T_{10}}{2} - T_A\right)$$

COMMON CASE DRAIN LINE

$$KQ_C C_p T_{11} = KQ_C C_p T_{16} + hA_C\left(\frac{T_{11} + T_{16}}{2} - T_A\right) + BTUHE$$

8000 PSI PUMP SUCTION LINE

$$KQ_{S8} C_p T_{13} = KQ_{S8} C_p T_{14} + hA_{SUCT8}\left(\frac{T_{13} + T_{14}}{2} - T_A\right)$$

3000 PSI PUMP SUCTION LINE

$$KQ_{S3} C_p T_{13} = KQ_{S3} C_p T_{15} + hA_{SUCT3}\left(\frac{T_{13} + T_{15}}{2} - T_A\right)$$

COMMON SUCTION

$$KQ_S C_p T_{12} = KQ_S C_p T_{13} + hA_{SUCT}\left(\frac{T_{12} + T_{13}}{2} - T_A\right)$$

CASE DRAIN LINE JUNCTION

$$Q_C T_{11} = Q_{C3} T_8 + Q_{C8} T_{10} + Q_C T_{16}$$

SYSTEM RETURN LINE JUNCTION

$$Q_R T_5 = Q_{D3} T_2 + Q_{D8} T_4$$

8000 PSI PUMP OUTLET

$$T_3 = T_{14} + 16$$

3000 PSI PUMP OUTLET

$$T_1 = T_{15} + 9$$

ELEVATOR ACTUATOR

$$KQ_{D3} C_p T_{18} - KQ_{D3} C_p T_{17} = BTUE$$

3000 PSI SYSTEM RETURN LINE

$$KQ_{D3} C_p T_{18} = KQ_{D3} C_p T_2 + hA_{R3}\left(\frac{T_2 + T_{18}}{2} - T_A\right)$$

AILERON ACTUATOR

$$KQ_{D8} C_p T_{20} - KQ_{D8} C_p T_{19} = BTUA$$

8000 PSI SYSTEM RETURN LINE

$$KQ_{D8} C_p T_{20} = KQ_{D8} C_p T_4 + hA_{R8}\left(\frac{T_4 + T_{20}}{2} - T_A\right)$$

AND:

$Q_{S8} = Q_{D8} + Q_{C8}$

$Q_{S3} = Q_{D3} + Q_{C3}$

$Q_S = Q_{D3} + Q_{D8} + Q_C$

$Q_C = Q_{C3} + Q_{C8}$

[illegible] $= Q_C$

$BTUA = 1.486\ Q_{D8} P_8$

$BTUE = 1.486\ Q_{D3} P_3$

FIGURE 6 THERMAL ANALYSIS EQUATIONS

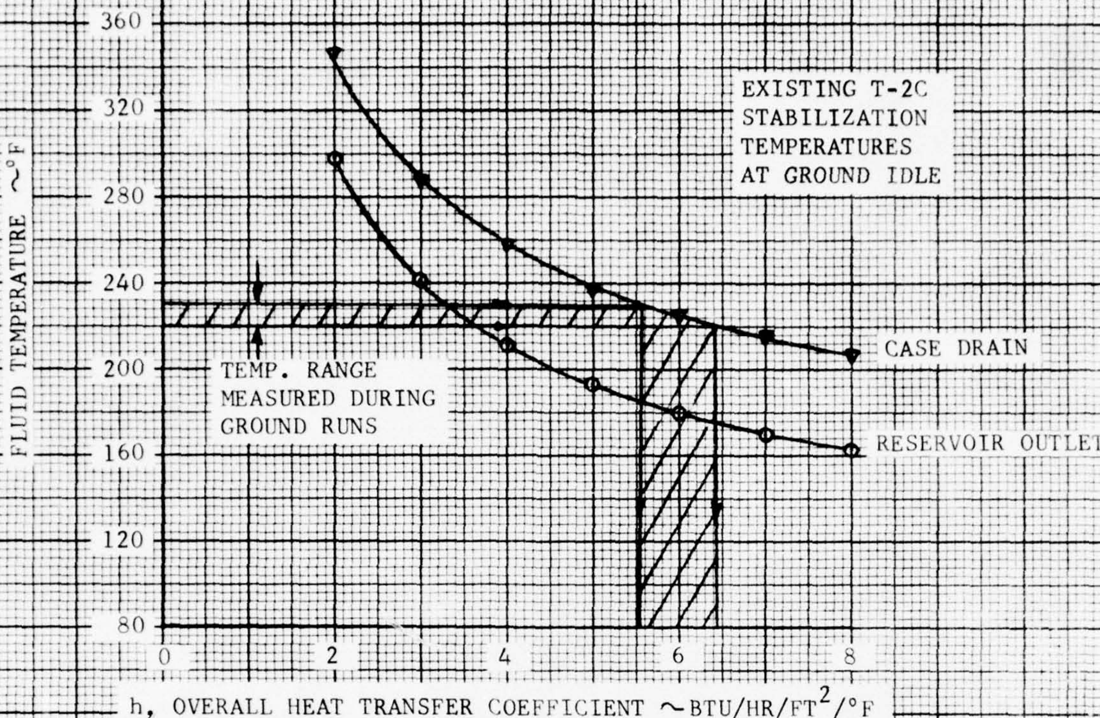


FIGURE 7 EXISTING T-2C SYSTEM TEMPERATURES AT GROUND IDLE

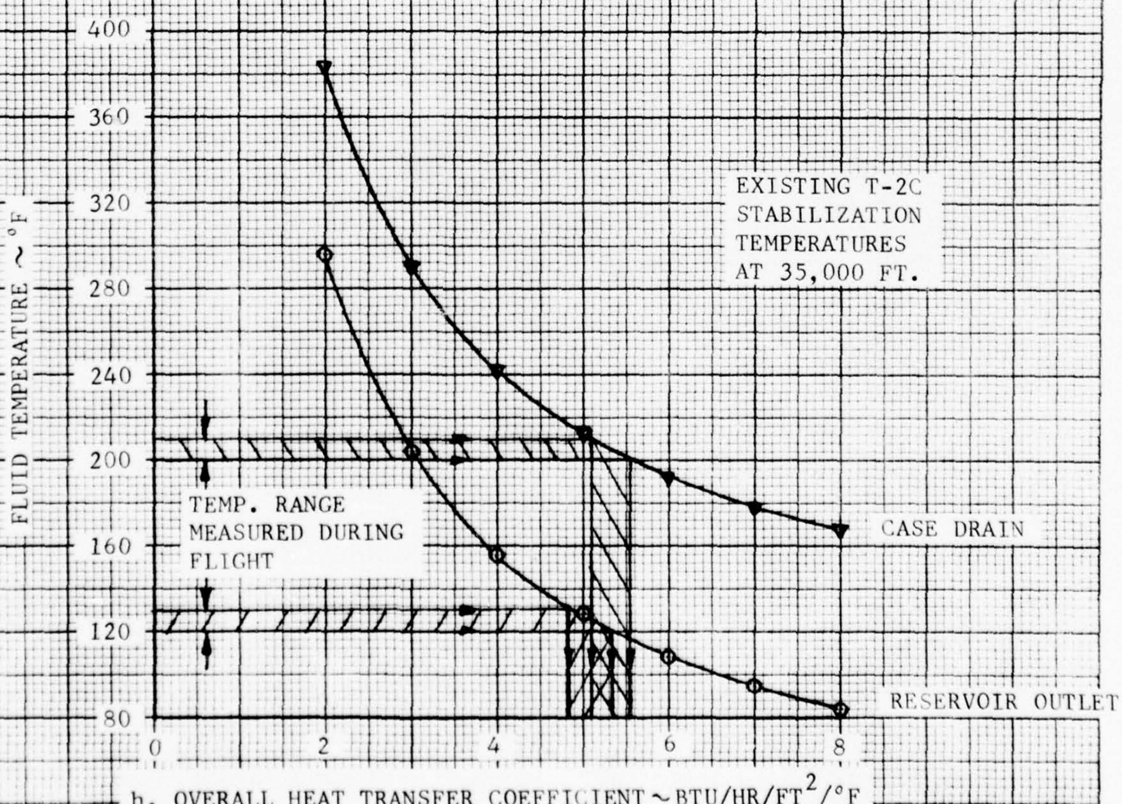


FIGURE 8 EXISTING T-2C SYSTEM TEMPERATURES AT 35,000 FT.

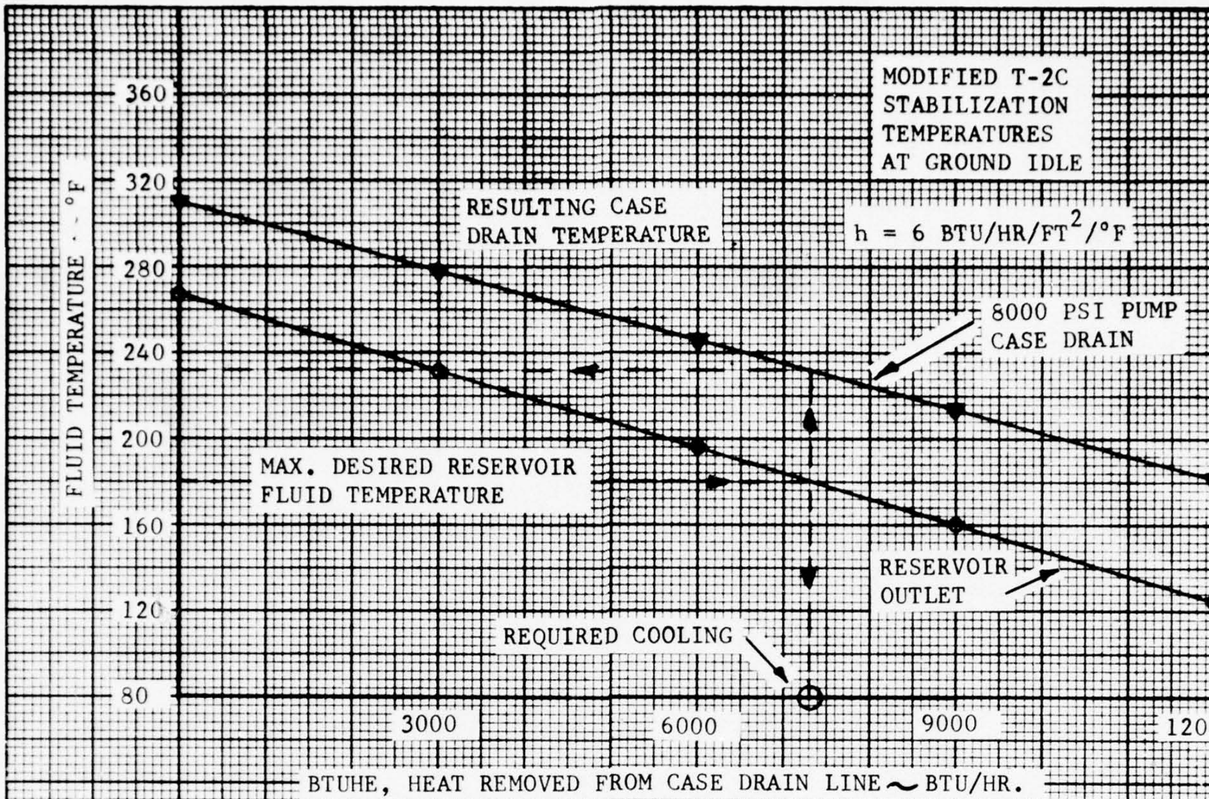


FIGURE 9 MODIFIED T-2C SYSTEM TEMPERATURES AT GROUND IDLE

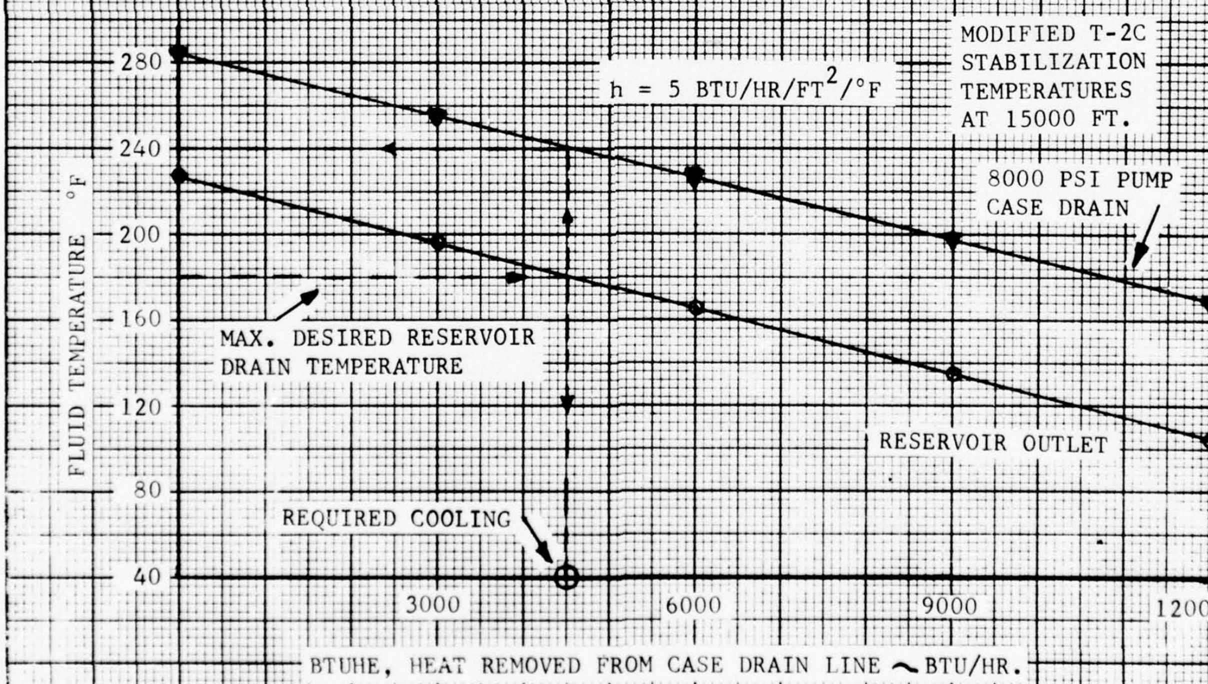


FIGURE 10 MODIFIED T-2C SYSTEM TEMPERATURES AT 15,000 FT.

2.4 SUCTION LINE ANALYSIS

2.4.1 Line Routing

The modified system suction line configuration is essentially the same as the original T-2C suction line except for minor changes at the 8000 psi pump. The two pumps (3000 psi and 8000 psi) will utilize a common 3/4 in. O.D. line from the reservoir which splits into two 5/8 in. O.D. lines -- one going to each pump. Line sizes, lengths, and fittings used in the modified system are shown on Figure 11.

2.4.2 Flow Conditions

Flow conditions examined were: (1) normal steady state flow (primarily system internal leakage); (2) maximum instantaneous (peak) flow; and (3) maximum continuous flow. Individual subsystem flow rates for the 3000 psi and 8000 psi systems are listed below:

	FLOW, GPM	
	STEADY STATE	MAXIMUM (PEAK)
3000 PSI SYSTEM		
Speed brake actuator	0	3.5
Landing Gear	0	4.5
Elevator Actuator	0.04	0.6
Pump Case Drain	0.20	0.2
System Internal Leakage	0.12	0.12
8000 PSI SYSTEM		
Aileron Actuator	0.03	0.33
Actuator Dynamic Seal Leakage (Internal)	0.06	0.04
Pump Case Drain	0.82	0.78

Based on the above, the following flow rates were used in the suction line analysis:

	SUCTION LINE FLOW, GPM	
SYSTEM	STEADY STATE	MAXIMUM (PEAK)
3000 PSI	0.36	4.70*
8000 PSI	0.91	1.15

*Rated delivery of 3000 psi pump plus case drain flow.

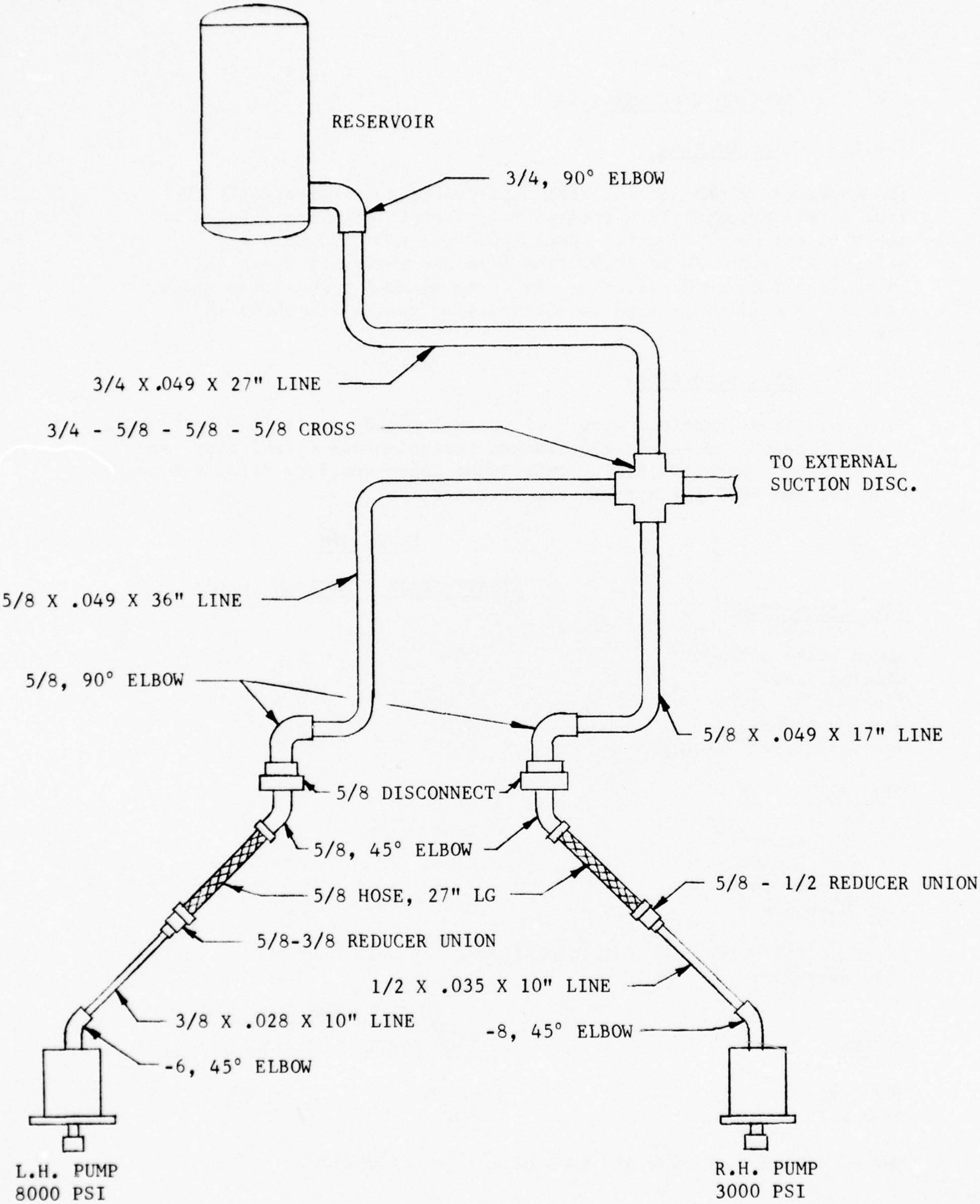


FIGURE 11 MODIFIED T-2C SYSTEM SUCTION LINE SCHEMATIC

2.4.3 Calculations

Sufficient pressure must be provided at the inlet port to prevent pump induced fluid cavitation. Suction pressure is a function of several parameters: reservoir pressure, fluid column head, line losses, and dynamic response of the system. Adequate inlet pressure must not only be maintained while the pump is delivering rated flow, but must also be maintained during rapidly changing flow conditions -- in particular the steady state flow to full flow condition. Assumptions made and the procedure used to determine available suction port pressure were as follows:

Fluid Temperature - A fluid temperature of +120°F was assumed. Actual inflight temperatures will be somewhat higher resulting in less suction line pressure loss. Thus, +120°F is a conservative value.

Flight Altitude - A maximum altitude of 20,000 ft. was assumed. Ambient pressure is therefore 6.75 PSIA.

Reservoir Pressure - Air pressure in the reservoir is regulated to 20 $^{+5}_{-0}$, psi above ambient pressure. Minimum reservoir pressure at 20,000 ft. is therefore 20 + 6.75 = 26.75 PSIA.

Line Losses - Total pressure loss in the suction line is equal to the sum of the losses in the line elements. Pressure loss in each element is given on Table III. These values were established using viscosity data from Reference 10 and empirically derived pressure loss charts covering tubing and fittings developed at CAD.

Fluid Column Head - The reservoir fluid level is normally above the pump suction port. The distance varies with the attitude of the aircraft, but 8 inches is a representative figure. A positive head of 0.24 psi due to fluid column weight was therefore assumed.

Flow Acceleration Pressure - The pump delivery control mechanism can regulate between rated discharge pressure and maximum full flow pressure within 0.050 sec. Fluid flow in the suction line must therefore accelerate from steady state flow (system internal leakage) to maximum flow within 0.050 sec. The pressure required to accelerate a column of fluid is:

TABLE III SUCTION LINE PRESSURE LOSSES AT +120°F
(REFERENCE FIGURE 11)

ITEM	STEADY STATE ΔP(PSI)	STEADY STATE FLOW(GPM)	MAXIMUM FLOW ΔP(PSI)	MAXIMUM FLOW FLOW(GPM)
COMMON LINE				
3/4, 90° ELBOW	.02		.1	
3/4 X .049 X 27 LINE	.07	1.27	.34	5.85
3/4-5/8-5/8-5/8 CROSS	.02		.4	
TOTAL	.11		.84	
8000 PSI PUMP LINE				
5/8 X .049 X 36 LINE	.03		.05	
5/8, 90° ELBOW	.02		.02	
5/8, DISCONNECT	.02		.02	
5/8, 45° ELBOW	.02	.91	.02	1.15
5/8 HOSE, 27" LG	.03		.03	
5/8-1/2 REDUCER UNION	.02		.02	
1/2 X .035 X 10 LINE	.07		.09	
1/2, 45° ELBOW	.10		.10	
TOTAL	.31		.35	
3000 PSI PUMP LINE				
5/8 X .049 X 17 LINE	.02		.37	
5/8, 90° ELBOW	.02		.4	
5/8 DISCONNECT	.02		2.0	
5/8, 45° ELBOW	.02	.36	.6	4.7
5/8 HOSE, 27" LG	.03		.6	
5/8-1/2 REDUCER UNION	.02		.5	
1/2 X .035 X 10 LINE	.03		.79	
1/2, 45° ELBOW	.02		1.0	
TOTAL	.18		6.26	
SUMMARY				
ΔP - 3000 PSI PUMP	.29		7.10	
ΔP - 8000 PSI PUMP	.42		1.19	

$$P = \frac{W L \Delta V}{144 \, gt}$$

where, P = Flow acceleration pressure, psi

W = Density of fluid, lb/ft^3 (=52.7)

L = Length of fluid column, ft.

ΔV = Change in fluid flow velocity, ft/sec

g = Acceleration due to gravity, ft/sec^2 (=32.2)

t = Time interval, sec (=0.050)

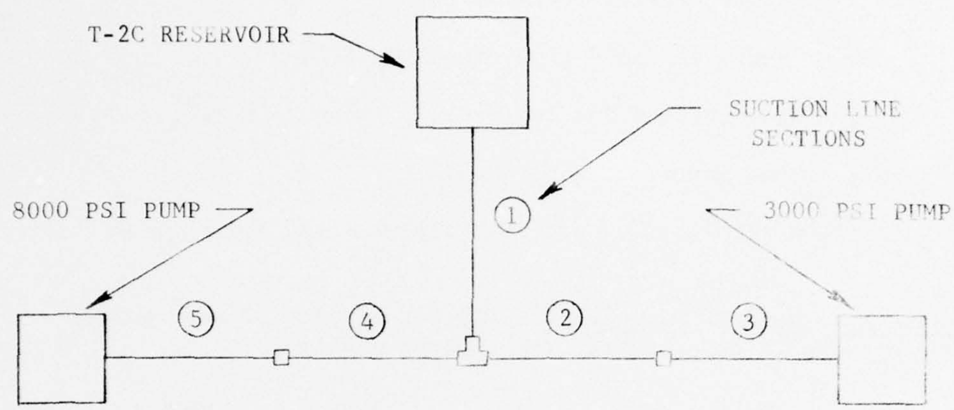
Results of flow acceleration pressure calculations are presented in Table IV.

Pressure Available at Pump Inlet - The net results of the factors affecting pump inlet pressure are summarized below:

Steady State Flow	3000 psi Pump	8000 psi Pump
Minimum reservoir pressure at 20,000 ft.	26.75 psia	26.75 psia
Suction line Δp	-0.29	-0.42
Static head	+0.24	+0.24
Pressure available at pump inlet	26.70	26.57
Maximum Continuous Flow		
Minimum reservoir pressure at 20,000 ft.	26.75	26.75
Suction line Δp	-7.10	-1.19
Static head	+0.24	+0.24
Pressure available at pump inlet	19.89	25.80
Maximum Peak Flow		
Minimum reservoir pressure at 20,000 ft.	26.75	26.75
Suction line Δp (average)	-3.70	-0.81
Static head	+0.24	+0.24
Flow acceleration pressure	-8.98	-0.91
Pressure available at pump inlet	14.31	25.27
Minimum pressure required at pump inlet based on test data	13.0	14.6

TABLE IV

PRESSURE REQUIRED TO ACCELERATE FLUID FROM STEADY STATE TO MAXIMUM FLOW



LINE	FLOW RANGE, GPM	VELOCITY FT/SEC	LENGTH, FT	PRESSURE, PSI
	8000 PSI PUMP			
1	.9-1.2	.29	2.25	.15
4	.9-1.2	.44	5.25	.53
5	.9-1.2	1.20	.83	.23
	FLOW ACCELERATION PRESSURE		=	.91 PSI
	3000 PSI PUMP			
1	.4-4.7	4.03	2.25	2.07
2	.4-4.7	6.18	3.67	5.16
3	.4-4.7	9.28	.83	1.75
	FLOW ACCELERATION PRESSURE		=	8.98 PSI

The analysis shows that the smallest margin between required and anticipated inlet pressures occurs during the maximum peak flow condition when the pump is rapidly responding to a sudden increase in demand from steady state to maximum flow. The margin for the 8000 psi pump is more than adequate for all flight conditions. Suction port pressure for the 3000 psi pump is low, however the actual margin will be somewhat higher since a minimum reservoir pressure was assumed. The small margin is due to the fact that the T-2C 3000 psi pumps were originally sized to meet maximum system demands with both pumps operating. The modified system has only one 3000 psi pump (and one 8000 psi pump). The margin is normally 5 psi with two 3000 psi pumps. The 1.31 psi margin for the test installation is considered acceptable provided flight testing is conducted below 20,000 ft altitude.

It should be noted that the foregoing analysis was conducted using early data (from Abex) on the 8000 psi pump. Subsequent changes made to the pump resulted in lower case drain flow. The performance improvement reduces steady state suction line flow, consequently, suction line pressure loss is also reduced. The steady state portion of the analysis is therefore conservative.

3.0 PUMP TESTS

3.1 DESCRIPTION

The LHS pump was designed and fabricated by the Aerospace Division of Abex Corporation in Oxnard, California, Reference 8. The unit was built under Contracts N62269-74-C-0798 and N62269-75-C-0300 to support the Lightweight Hydraulic System and Advanced Flight Control Actuation System R&D programs sponsored by the Naval Air Development Center, Reference 9. Technical consultation required during pump development was provided by CAD. The unit was configured to mount on the T-2C L.H. engine auxiliary power gearbox. Pump operating speeds are:

Engine Idle:	3660 rpm
Cruise:	7330 rpm
100% Power:	7800 rpm

The 8000 psi pump, identified as M/N APIV-106, P/N 63077, S/N 151353, is a constant pressure, variable displacement axial piston design, Figure 12. Rated delivery is 3 gpm at 7330 rpm and 7850 psi with a +240°F inlet temperature. Theoretical displacement is 0.124 CIPR. The pump weighs 7.5 pounds (wet). Port sizes are: -6 inlet, -6 discharge, and -4 case drain.

The pump has 9 pistons in a rotating cylinder barrel. Piston shoes slide on an inclined cam face causing the pistons to reciprocate as the barrel rotates. One end of the barrel slides against a valving surface on a port plate through which fluid enters and is discharged. The cylinder barrel is supported by roller bearings sized to react force components developed by pressure on the pistons. Cylinder barrel thrust and piston shoe forces are hydrostatically balanced. Flow output is varied by means of a compensating-valve/stroking - piston mechanism. The stroking-piston controls the tilt angle of the cam hanger so that CIPR displacement matches system demand.

The cylinder barrel is 4340 steel heat treated to 30 minimum Rockwell C; the port face is plated with shoe bronze. The piston bores in the barrel contain Mueller bronze sleeves. Maximum working stress level in the barrel is estimated to be 25,000 psi; the ultimate strength is approximately 140,000 psi. Piston material is 52100 steel heat treated to 58 minimum Rockwell C.

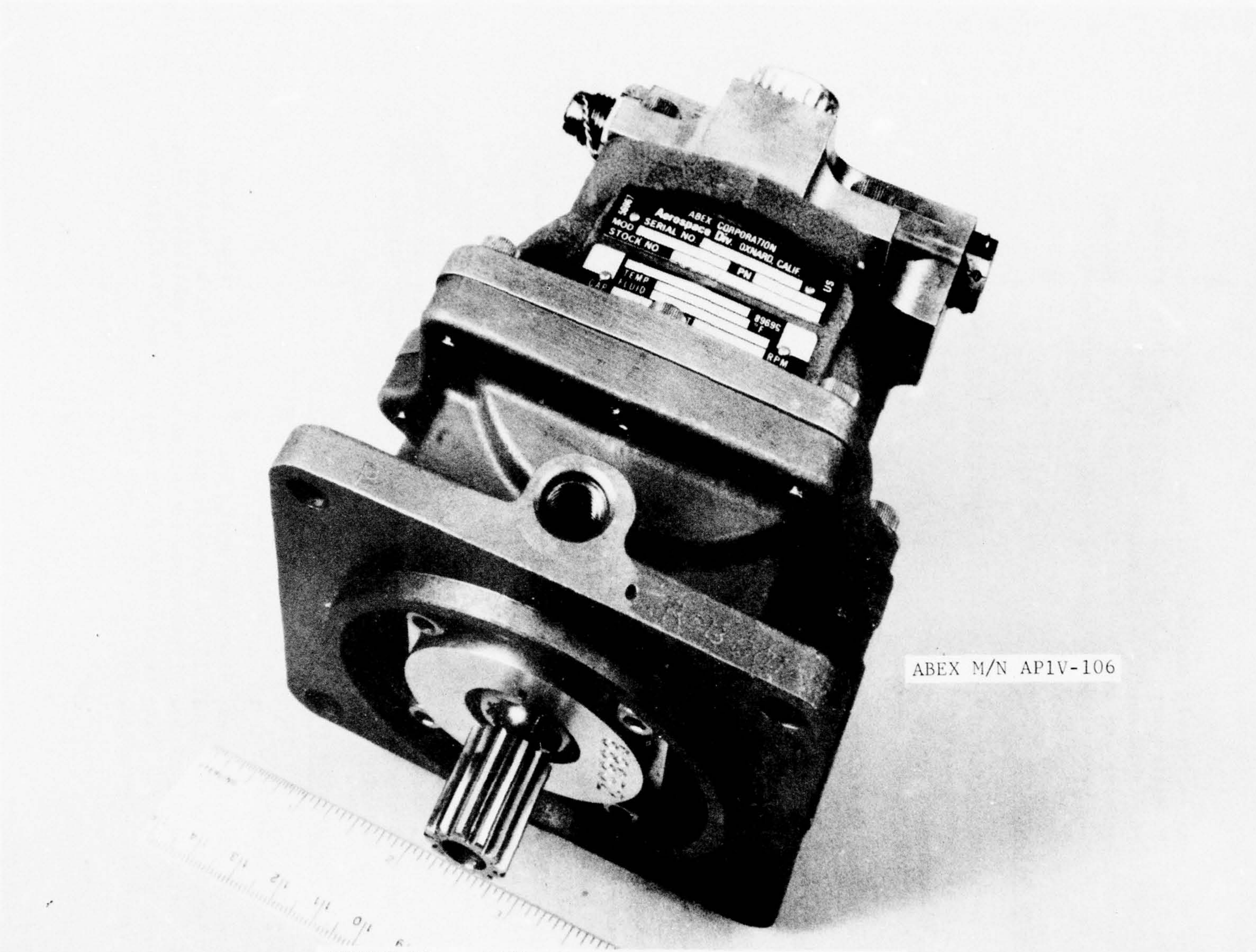


FIGURE 12   8000 PSI VARIABLE DISPLACEMENT PUMP

The LHS pump design was adapted from Abex M/N APIV-105 used on the Bell 214 Helicopter. M/N APIV-105 operates at 3000 psi, has a 0.310 CIPR displacement, and a rated speed of 8800 rpm. The major differences and simularities in the designs of the 8000 psi APIV-106 pump versus the 3000 psi APIV-105 are listed below:

Major Differences

Rotating barrel and piston group completely redesigned
Port cap re-designed
*Compensator spring changed
*Control valve changed

* These parts were "borrowed" from other Abex pump designs.

Major Similarities

Pump housing essentially unchanged
Shaft seal not changed
Bearing loads approximately the same
Shaft bearings not changed

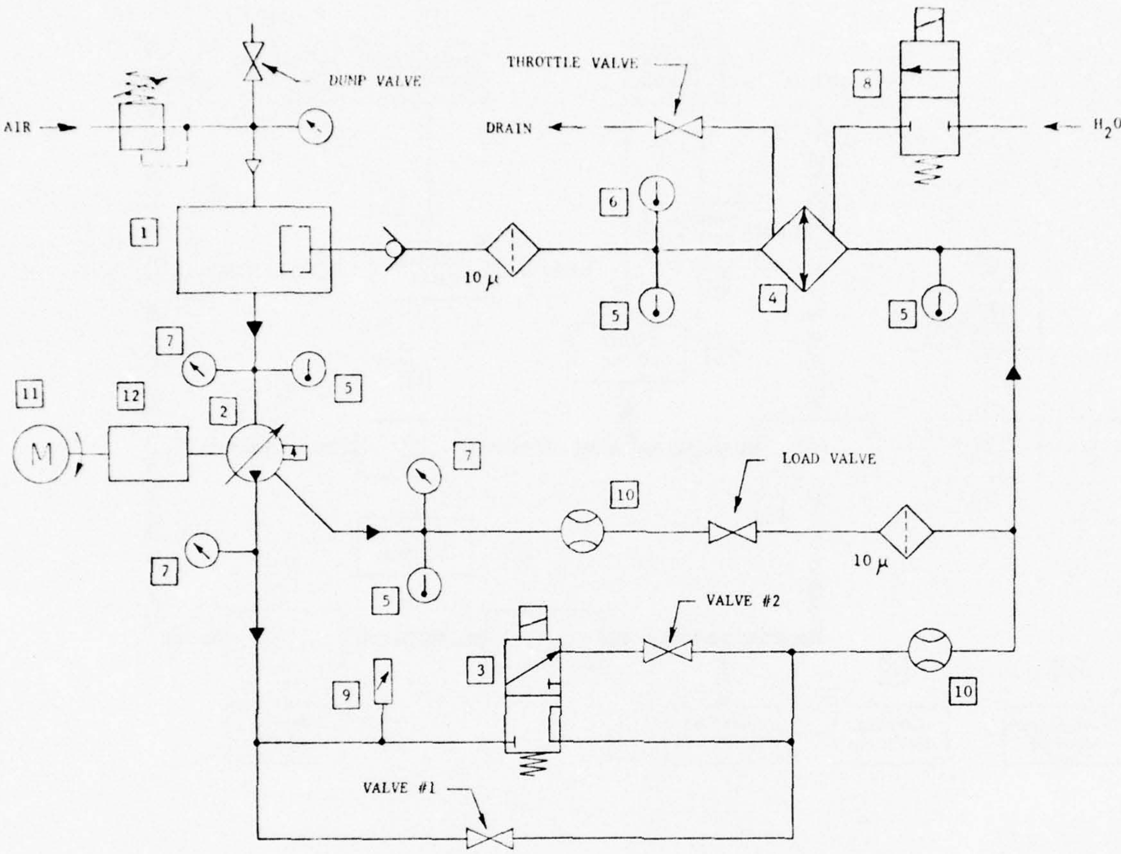
The 8000 psi pump contains no unique features. All parts in the pump are conventional components designed so that working stress levels are maintained within acceptable limits.

Since the 8000 psi lateral control system contains only one actuator, flow requirements are low. Pump flow capability at cruise speed is 3.2 gpm; actuator rated flow is 0.3 gpm. The pump is thus oversized. This results in a higher heat rejection than would be encountered using a smaller pump. Pump sizing was dictated by development costs, scheduling, and the capability of existing hardware which was modified. Future test programs are anticipated that will more fully utilize the LHS pump capacity.

3.2 PERFORMANCE TESTS

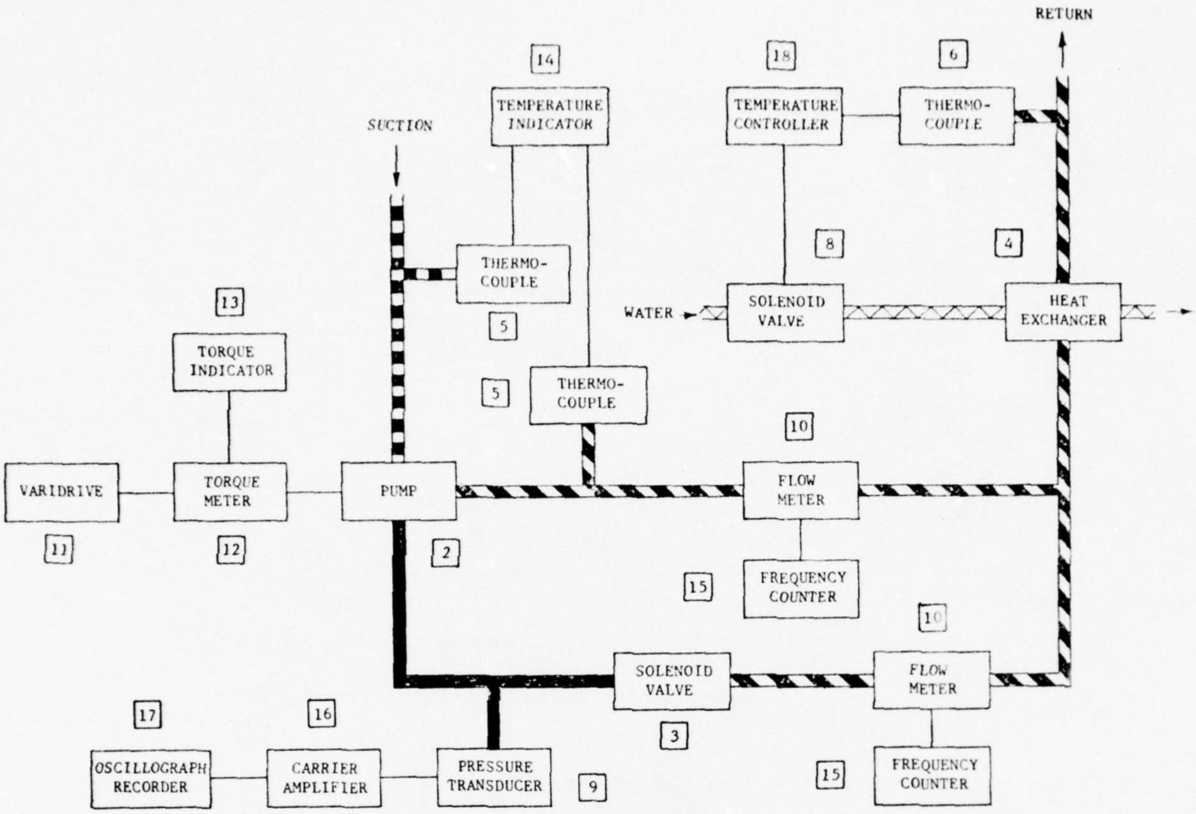
3.2.1 Efficiency

Overall efficiency was determined using the test system illustrated schematically on Figure 13. A block diagram of the instrumentation is shown on Figure 14. Testing was conducted with the solenoid valve de-energized and using needle valve #1 to control pump discharge flow, Figure 13. Parameters for the efficiency test were:



1	T-2C Hydraulic System Reservoir, air/oil type with air separator, Rockwell International P/N 288-580600-11
2	8000 psi pump, variable delivery, Abex M/N APIV-106
3	8000 psi Solenoid Valve, 3 port, Sterer Engineering P/N 15390-1
4	Heat Exchanger, Yates-American P/N RD-LL-4344-10
5	Copper-Constantan Thermocouple, Rockwell International P/N none
6	Iron-Constantan Thermocouple, Rockwell International P/N none
7	Pressure gage, bourden tube, Duragauge P/N none
8	150 psi Solenoid Valve, 2 port, Skinner Electric P/N LC2DB4150
	See Figure 14 for instrumentation

FIGURE 13 SCHEMATIC OF PUMP TEST HYDRAULIC SYSTEM



See Figure 13 for hydraulic equipment

9 Pressure Transducer, 15,000 psi, Standard Controls M/N 211-35-000

10 Turbine Flowmeter, .2 to 10 gpm, Fischer & Porter M/N 10C1510A

11 Varidrive, 75 hp, U.S. Electric Motors Type VEU-GSDT

12 Torque Meter, 1000 lb-in, B&F Instruments M/N 1000CB3

13 Torque Indicator, B&F Instruments M/N 1480-11

14 Temperature Indicator, 12 points, Brown Electronic M/N 156X63P12

15 Frequency Counter, Beckman/Berkeley M/N 7370

16 Carrier Amplifier, 4 Channels, Consolidated Electrodynamics Corporation M/N 1-127

17 Oscillograph Recorder, light beam type, direct writing, Minneapolis-Honeywell M/N 1108

18 Temperature Controller, Leeds & Northrup Series 60

FIGURE 14 BLOCK DIAGRAM OF PUMP TEST INSTRUMENTATION

Pump compensator setting: 8000 psig

Pump speed: 3660, 5400, 7330, 7800 rpm

Discharge pressure: 4000 to 8000 psig

Inlet fluid temperature: +110, +180°F

Reservoir pressure: 30 psig

Pump case drain pressure: 45 to 70 psig (observed)

Discharge fluid temperature: Recorded

Case drain fluid temperature: Recorded

Input torque: Recorded

Case drain flow: Recorded

Discharge flow: Recorded

Pump performance is summarized on Table V; efficiency curves are shown on Figure 15. Overall efficiency depended on operating conditions and averaged 89% (at 7500 psig). This value is nominal considering the pump type, size, and test conditions, and compares favorably with the efficiency level of conventional 3000 psi aircraft type pumps.

3.2.2 Response

Transient response tests were conducted with plumbing circuitry patterned after the requirements of paragraph 4.3.2.1.5.3 in MIL-P-19692B. Pump discharge flow was varied from 5% to 90% to 5% of full discharge flow by means of a solenoid valve and pre-set load valves, while an oscillographic record was made of the pressure/time function. Needle valve #1 was used for 5% flow; valve #2 was set to obtain 90% flow, Figure 13. The pressure fluctuations recorded were immediately upstream of the solenoid valve, Figures 16 and 17.

The peak surge observed following closure of the solenoid valve was 113% of rated system pressure (at +180°F). This is less than the 120% allowable of Reference 4, and the 135% allowable of MIL-P-19692B.

Response and stabilization time requirements of MIL-P-19692B are 0.050 sec (max.) and 1.0 sec (max.), respectively. Pump stabilization time was satisfactory; response time for the 5% to 90% flow condition exceeded the 0.050 sec limit. The rated speed of the 3000 psi M/N APIV-105 pump is 8800 rpm, Section 3.1. If the 8000 psi M/N APIV-106 pump were operated at 8800 rpm, response time would probably be satisfactory (note the trend between 3660 and 7330 rpm). The response time of the 8000 psi pump will have no detrimental affect on the LHS flight evaluation program, since system flow requirements (0.3 gpm) are much less than the 90% flow condition (2.8 gpm) used in the transient response tests.

TABLE V

PUMP PERFORMANCE SUMMARY

INLET FLUID TEMP, °F	DISCH. PRESS., PSIG	PUMP SPEED, RPM	DISCH. FLOW, GPM	CASE FLOW, GPM	CASE DRAIN TEMP, °F	HEAT REJECT., BTU/MIN	OVERALL EFFICIENCY, %
+110°F	7500	3660	1.58	0.16	148	35	89.5
		7330	3.22	0.20	184	78	88.4
+180°F	7500	3660	1.49	0.30	212	46	85.7
		5400	2.26	0.31	222	64	86.7
		7330	3.17	0.30	239	72	89.1
		7800	3.36	0.32	235	88	87.6
+110°F	8000	3660	0	0.52	160	113	0
		7330	0	0.57	175	142	0
+180°F	8000	3660	0	0.72	225	143	0
		5400	0	0.73	231	167	0
		7330	0	0.75	236	179	0
		7800	0	0.76	237	195	0

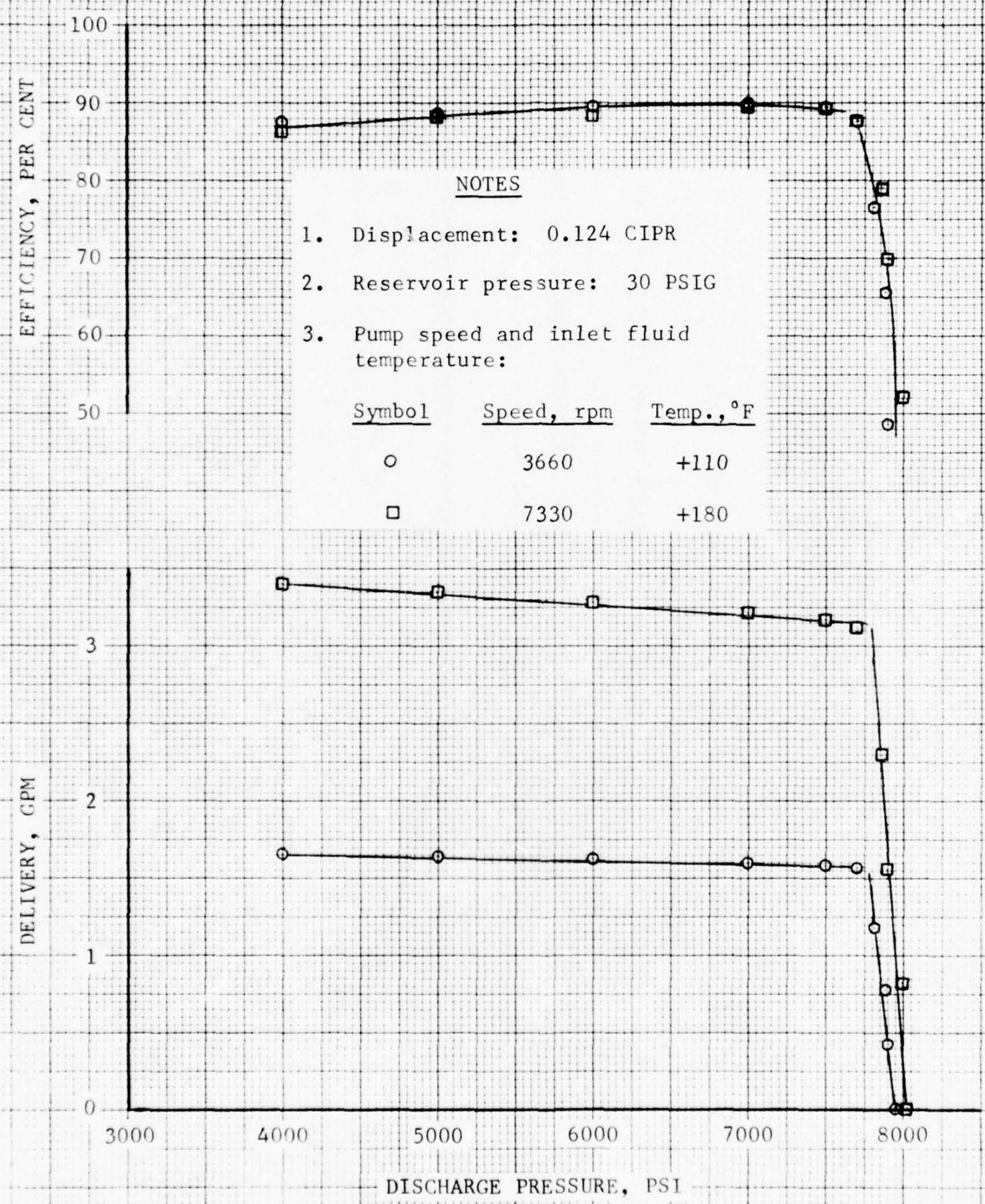
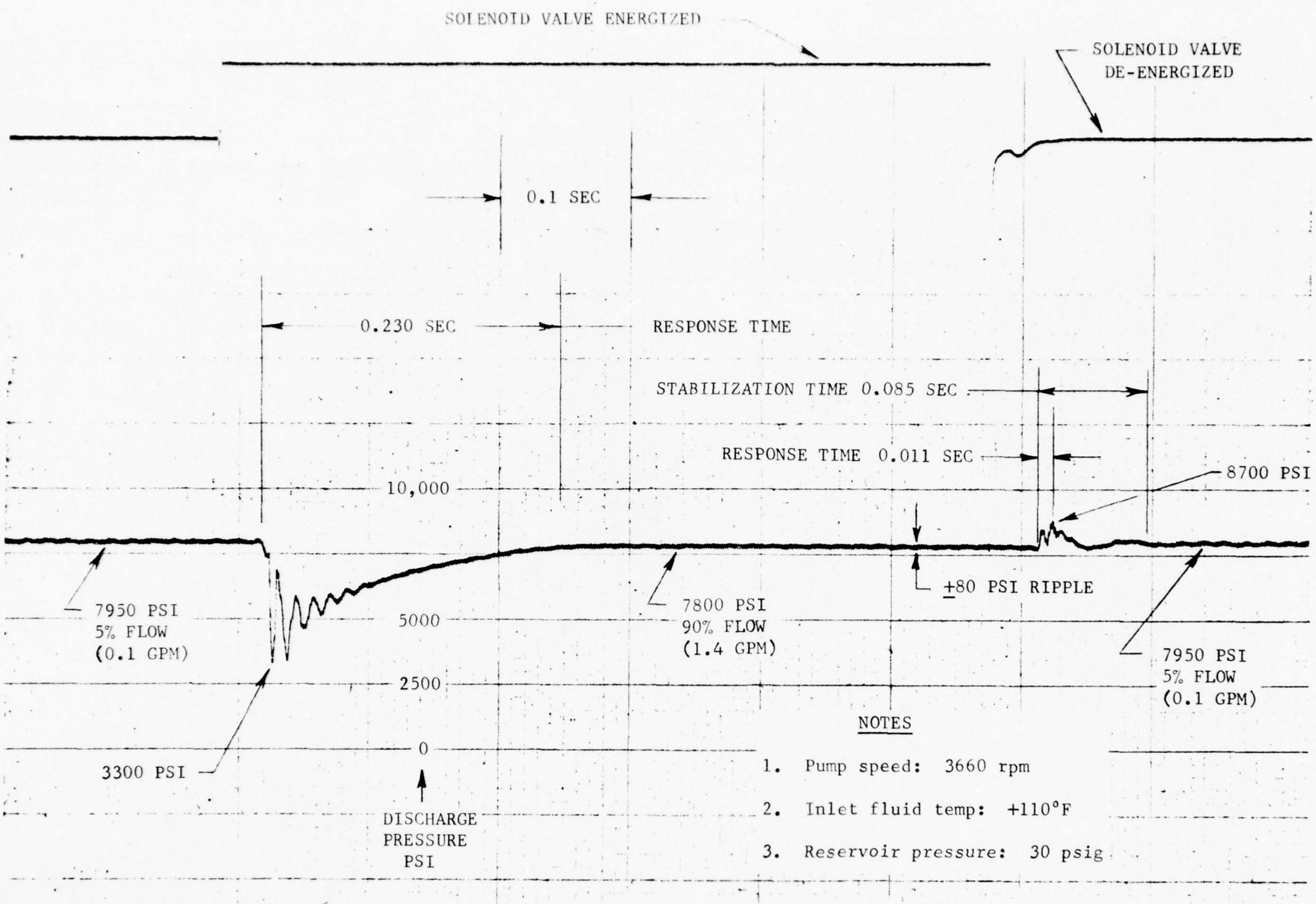


FIGURE 15 PUMP OVERALL EFFICIENCY



NOTES

1. Pump speed: 3660 rpm
2. Inlet fluid temp: +110°F
3. Reservoir pressure: 30 psig

FIGURE 16 PUMP TRANSIENT RESPONSE AT +110°F

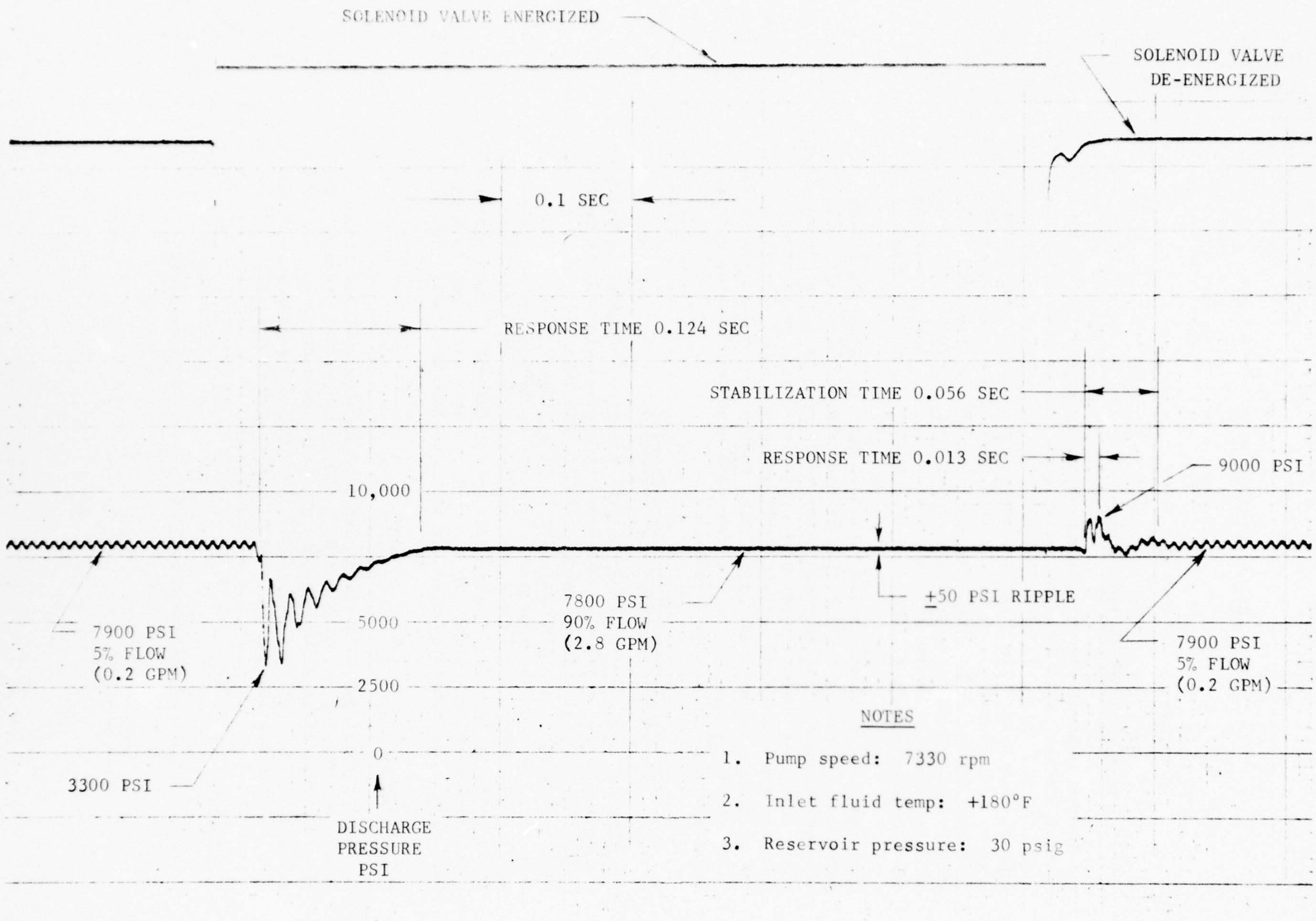


FIGURE 17 PUMP TRANSIENT RESPONSE AT +180°F

4.0 AILERON ACTUATOR TESTS

4.1 DESCRIPTION

4.1.1 T-2C Lateral Control System

The lateral control system consists of the pilot's stick, mechanical control linkage, trim actuator, artificial feel bungees (2), aileron actuator, and ailerons. The ailerons are incorporated in the trailing edge of each wing. The aileron actuator is a manual input, single system, full power, moving body design, and is located aft of the rear spar in the right wing, just outboard of the fuselage. The actuator piston rod is attached to the aircraft structure; the moving body is connected to the ailerons via mechanical linkage; and the actuator input lever is connected to push-pull control linkage.

When the pilot moves the stick, the actuator control valve opens, porting fluid to the desired side of the cylinder and positioning the ailerons by mechanical feedback. The actuator contains a bypass valve which functions during "power off" operations. This valve interconnects the extend and retract ports to permit manual control of the ailerons. With "power off", stick inputs move the control valve lever against stops (on the actuator) and push the actuator body to achieve manual aileron deflection.

4.1.2 LHS Aileron Actuator Design Details

The 8000 psi actuator built for flight testing is shown pictorially on Figure 18. This actuator was designed to replace (for test purposes only) the 3000 psi aileron servo actuator, P/N 249-58701, used in the T-2C. Outwardly, the 8000 psi unit is very similar to the 3000 psi actuator. Many of the same parts used on the 3000 psi assembly were used on the 8000 psi actuator. A cross-sectional view on Figure 19 shows internal features of the experimental actuator. Part numbers of components used exclusively in the 8000 psi unit are given on Figure 19. Major differences between the two actuators are listed below:

Body Forging

Production forging P/N 249-58731 was used for the 8000 psi actuator body. The forging was machined by a numerical control program. The program normally used for the 3000 psi body was revised for the 8000 psi body. The revisions included: 1) reduced cylinder bore diameter, 2) internal porting changes, and 3) two-stage rod seals.

Piston Rod

Piston area was reduced and a metallic piston seal was used.

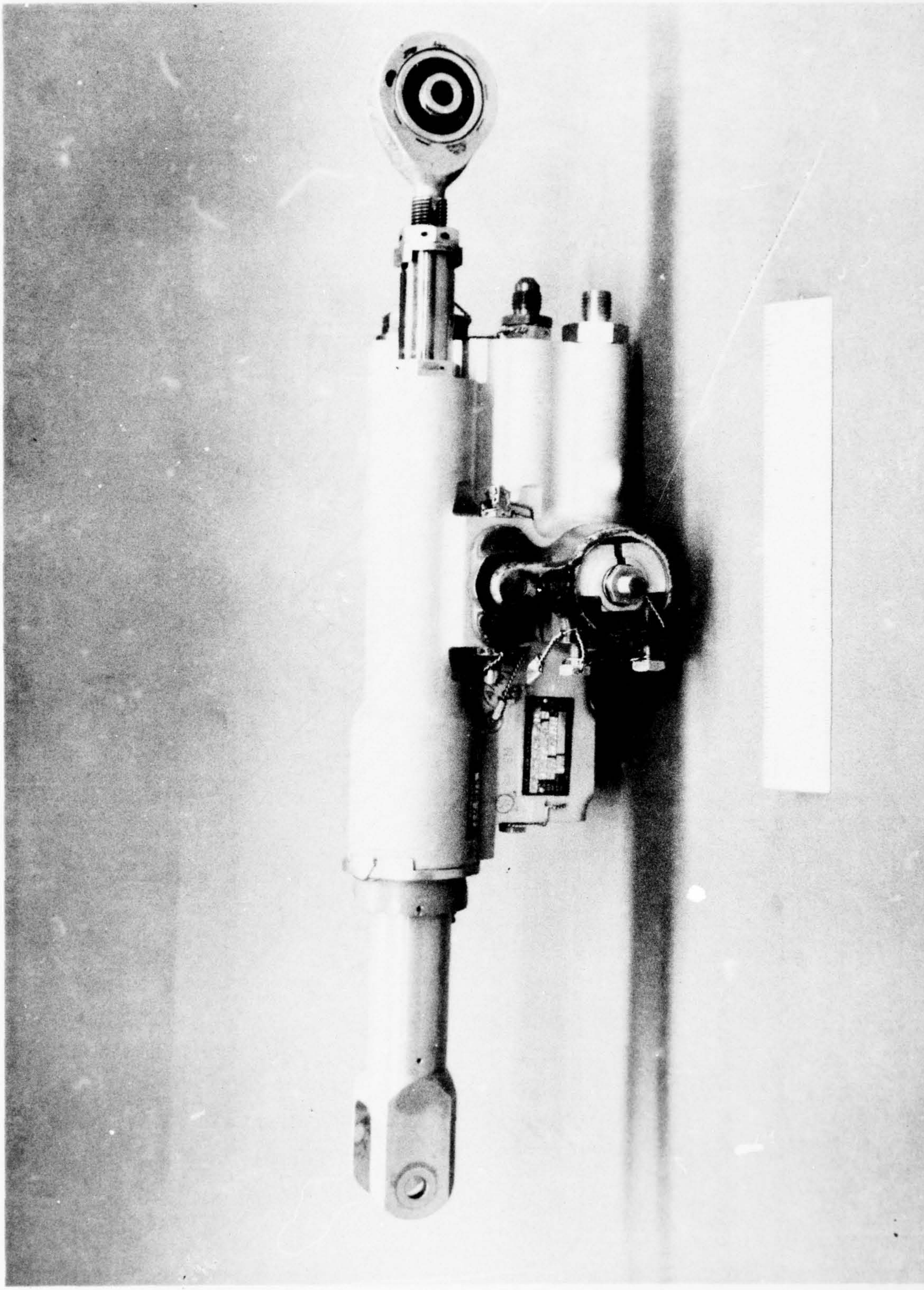


FIGURE 18 8000 PSI AILERON ACTUATOR

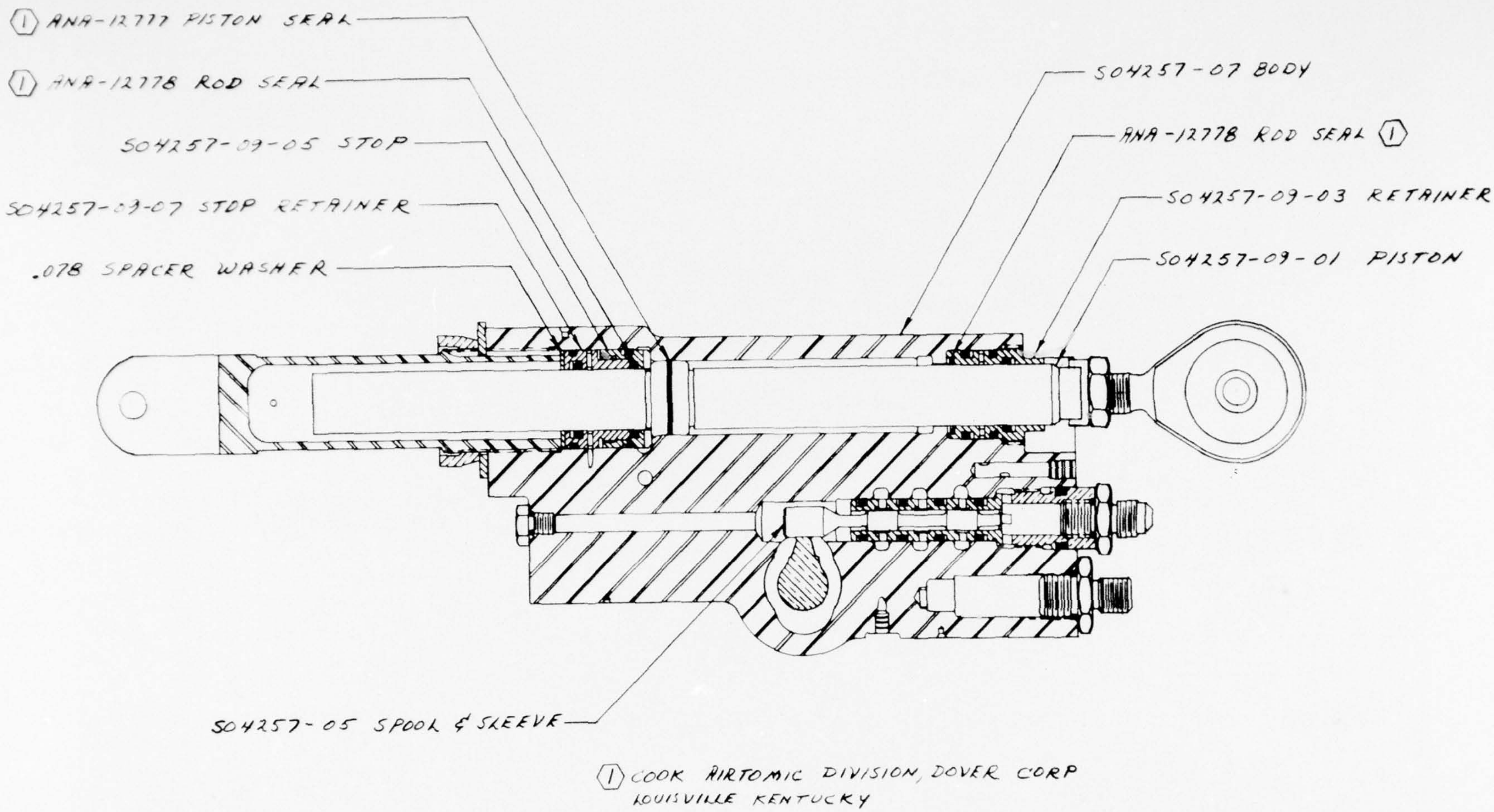


FIGURE 19 CROSS-SECTIONAL VIEW OF 8000 PSI AILERON ACTUATOR

Flow Control Valve

The control valve was designed for operation at 8000 psi.

Rod Seals

The rod seals are two-stage. The first stage is metallic; the second stage uses an MS28775 O-ring and MS28774 backups.

The LHS aileron actuator is an engineering model; no attempt was made to save weight by optimization, however efforts were made to minimize cost. MIL-C-5503 requirements were followed except as modified for the 8000 psi pressure level. Design constants of the actuator are summarized below:

Operating pressure	8000 psi
Piston stroke (total)	3.00 in.
Cylinder bore	0.926 in.
Cylinder wall thickness (min.)	0.256 in.
Rod diameter	0.748 in.
Piston effective area	0.234 $in.^2$
Force output (max.)	1872 lb
Piston velocity (max.)	5.4 in/sec
Actuator length (mid-stroke)	15.188 in.

The actuator body is a 2014-T6 aluminum die forging. The cylinder bore is hard anodized with a surface finish of 16-32 micro-inches roughness height. The piston/rod is fabricated from 4140 alloy steel and has a ground chrome plate surface with an 8-16 micro-inch finish.

Actuator piston travel and rate are commanded by a directional flow control valve. The valve spool is driven by a mechanical input lever on the side of the actuator, Figure 18.

4.1.3 Flow Control Valve

The control valve was designed by CAD and fabricated by Columbus Aircraft Nitriding Company, Columbus, Ohio, Figure 20. The valve is a spool/sleeve type, 4-way, proportional flow control design based on data developed in Reference 3. The spool is made of nitralloy and has a surface hardness of 90 minimum Rockwell 15N . Sleeve material is 52100 steel heat treated to 60 minimum Rockwell C. MS28775 O-rings

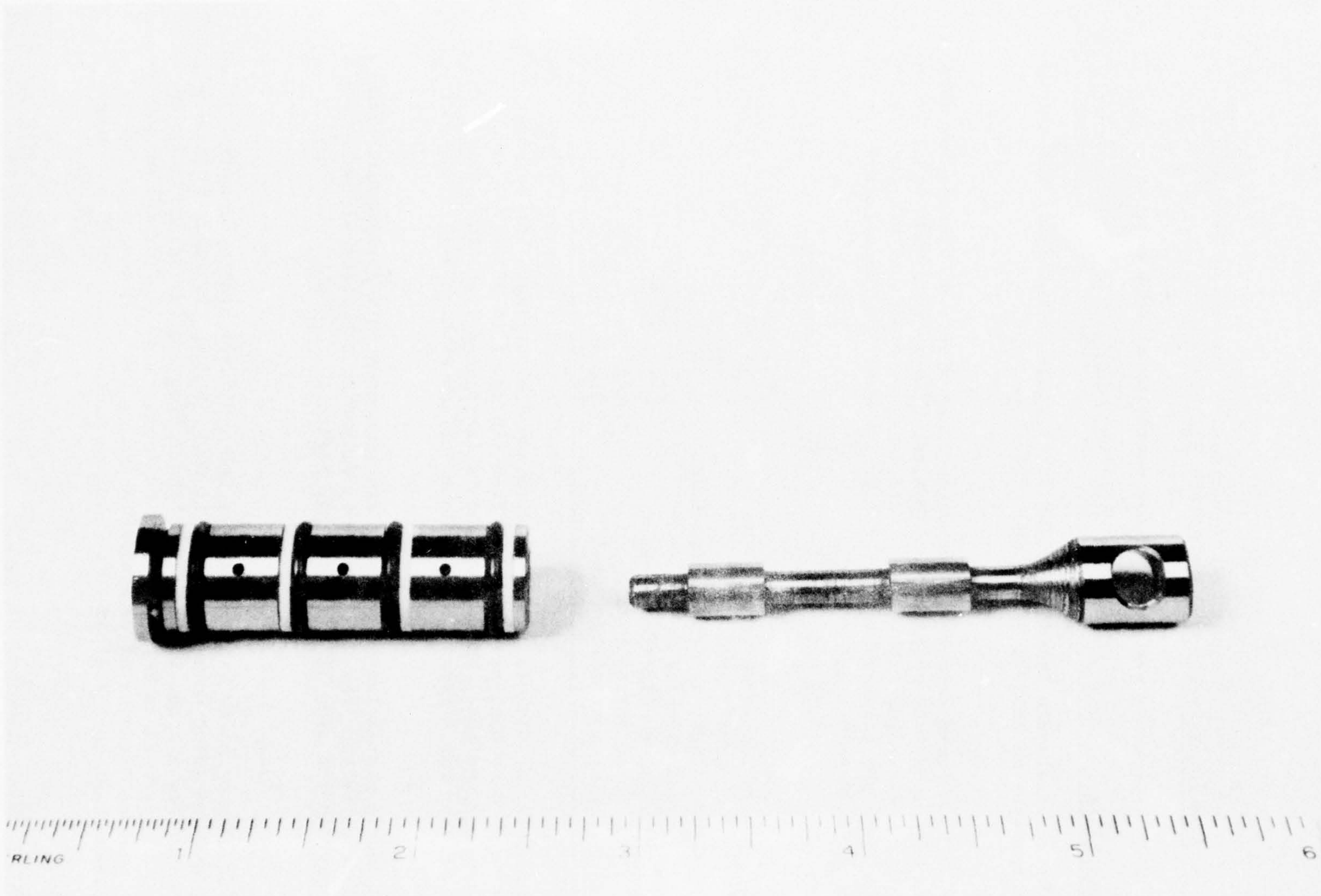


FIGURE 20 8000 PSI FLOW CONTROL VALVE

and MS 28774 backup rings are used in the static seals between the sleeve and actuator body. Pertinent physical characteristics are:

Spool type	2 land
Spool land diameter	0.250 in.
Spool travel (full)	±0.056 in.
Design overlap	∓0.003 in.
Notch type	60° "V" with 8° slope
Number of notches	3 notches/land edge (12 total)

Design performance requirements of the valve operating at 8000 psi with MIL-H-83282 fluid at +110°F are listed below:

Rated flow (at ±0.056 in.)	0.33 gpm
Internal leakage at null (max)	130 cc/min

4.2 SPOOL/SLEEVE PERFORMANCE

The valve assembly was installed in a test fixture which had a screw-spool-spring arrangement for controlling spool displacement. Spool/sleeve operating characteristics were determined using the setup shown schematically in Figure 21. A dial indicator sensed spool position, and a graduate and stop watch were used to measure flow rates. The valve was pressurized by a commercial hydraulic power supply with a maximum output of 0.21 gpm at 8000 psi. This was done for convenience and to conserve the life of the LHS pump. Spool/sleeve flow gain, pressure gain, and internal leakage tests were conducted. Inlet fluid temperature was +115 ± 5°F.

4.2.1 Flow Gain

Valve flow gain is shown on Figure 22. Flow at rated spool displacement of ±0.056 in. averaged 0.25 gpm. The design value was 0.33 gpm. This discrepancy will not affect actuator performance since input lever travel can be increased slightly to compensate for the difference. Valve dead band was approximately ±0.006 in.; the design goal was ±0.003 in. The 0.003 difference was attributed to the manufacturing process. Effects of the dead band on system performance were considered minimal, and did not warrant the cost

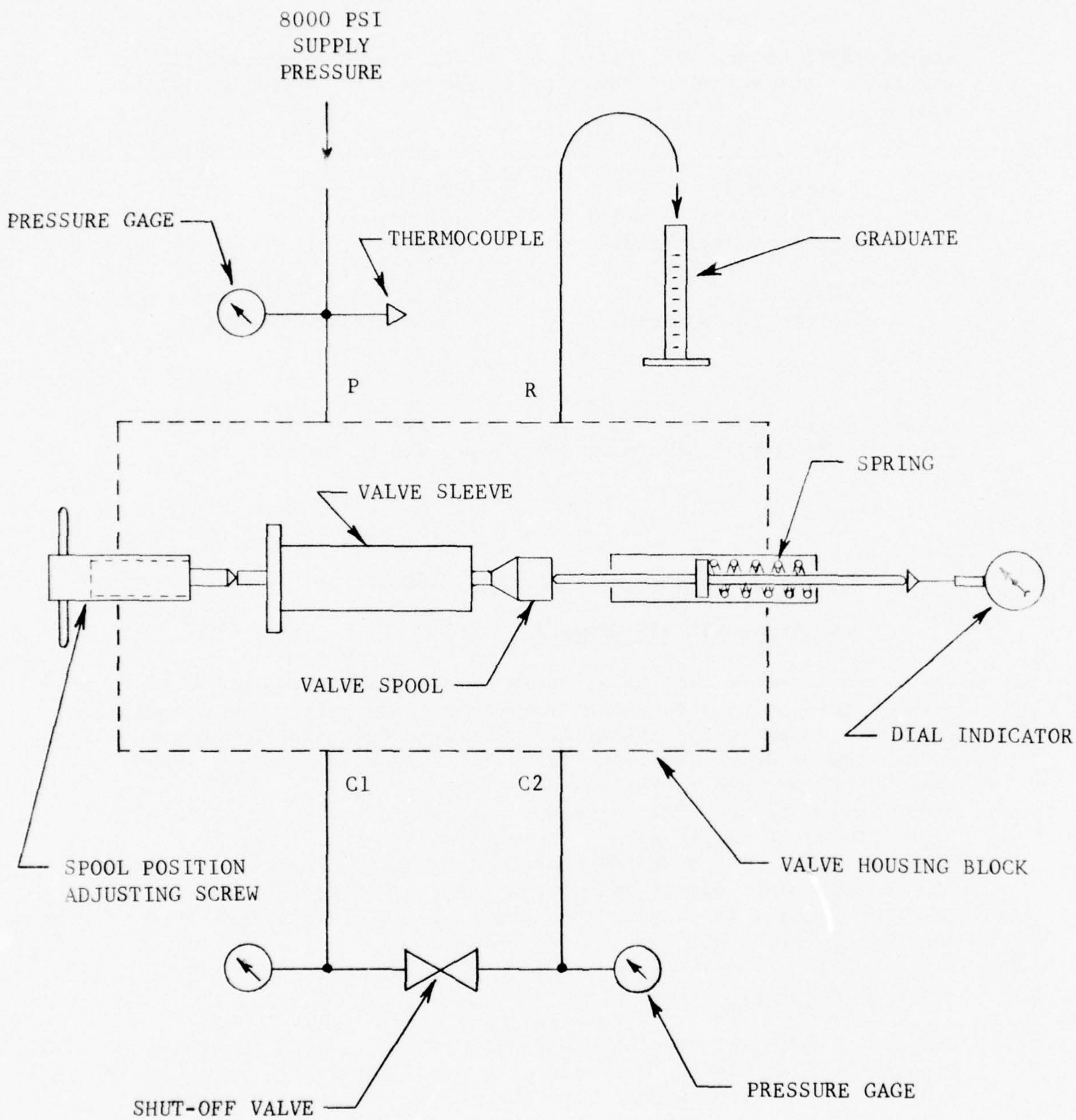


FIGURE 21 SCHEMATIC OF CONTROL VALVE TEST SETUP

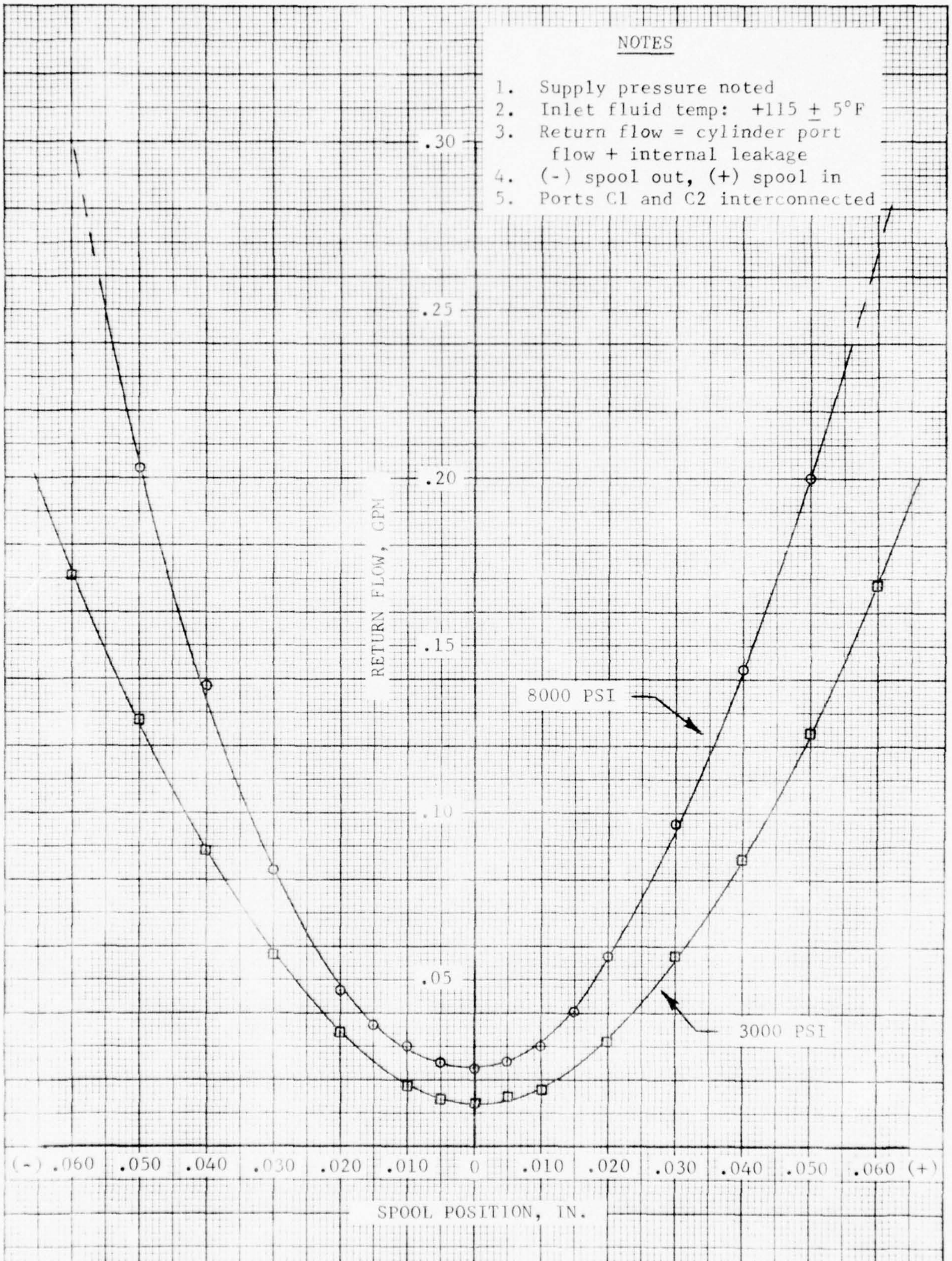


FIGURE 22 FLOW GAIN

of reworking the valve. Flow gain at 3000 psi is given on Figure 22 for reference purposes.

4.2.2 Pressure Gain

Pressure gain was approximately 7×10^5 psi/in, Figure 23. Values ranging around 2×10^6 psi/in are considered normal. Valve dead band reduced the pressure gain. High gains provide the actuator with the capability to break away large friction loads with little error. Since friction in the T-2C aileron system is relatively low, and since the pressure gain curve is shaped properly, performance degradation due to valve pressure gain is not anticipated.

4.2.3 Internal Leakage

The design objective of 130 cc/min maximum at +110°F was achieved. Peak leakage was 90 cc/min at null, Figure 24; null power loss was 0.11 hp. Valve dead band contributed to the excellent leakage characteristics.

4.3 OPERATIONAL TESTS

Actuator functional characteristics were determined and compared to requirements for the T-2C 3000 psi aileron actuator. A summary of this data is presented below:

	3000 psi Actuator Requirements	8000 psi Actuator Performance
Null Leakage, at +110°F, cc/min	210 (max)	125
Input lever dead band, in.	±0.005 (max)	±0.012
Full stroke time, sec	0.5 to 0.6 sec	0.55
Piston rod breakout friction, lb	12 (max)	12 to 18
Input lever operating force, lb	2 (max)	0.4

Internal leakage and piston rod friction in the 8000 psi actuator were both affected by the two-stage rod seals -- the 3000 psi unit has single-stage elastomeric rod seals. This should be considered in comparing null leakage and rod friction of the two actuators. The input lever dead band observed in the 8000 psi unit resulted from the spool/sleeve dead band noted in Section 4.2.1.

All pertinent operational characteristics of the unit were acceptable. The 8000 psi actuator was therefore considered satisfactory for evaluation testing in the T-2C lateral control system.

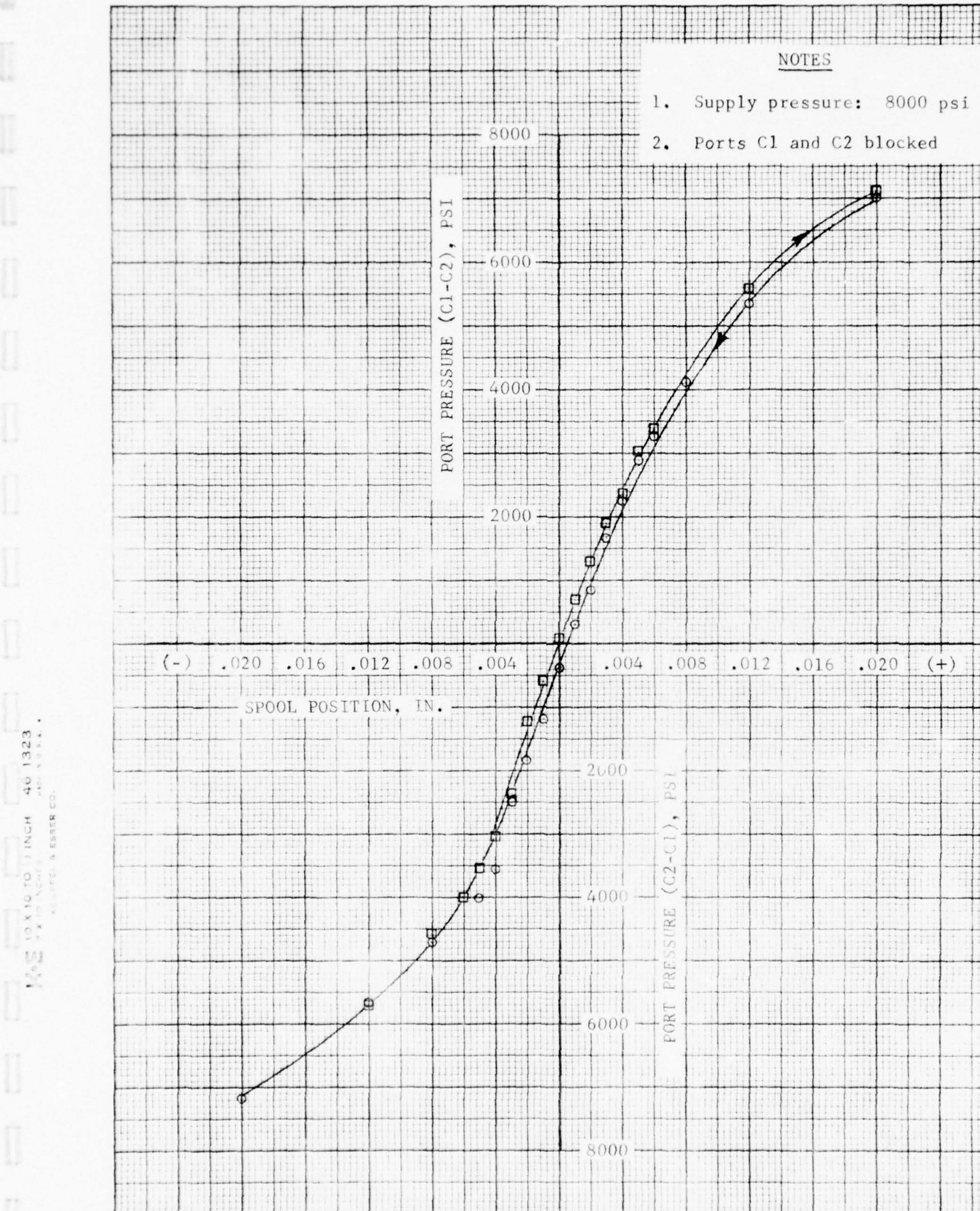


FIGURE 23 PRESSURE GAIN

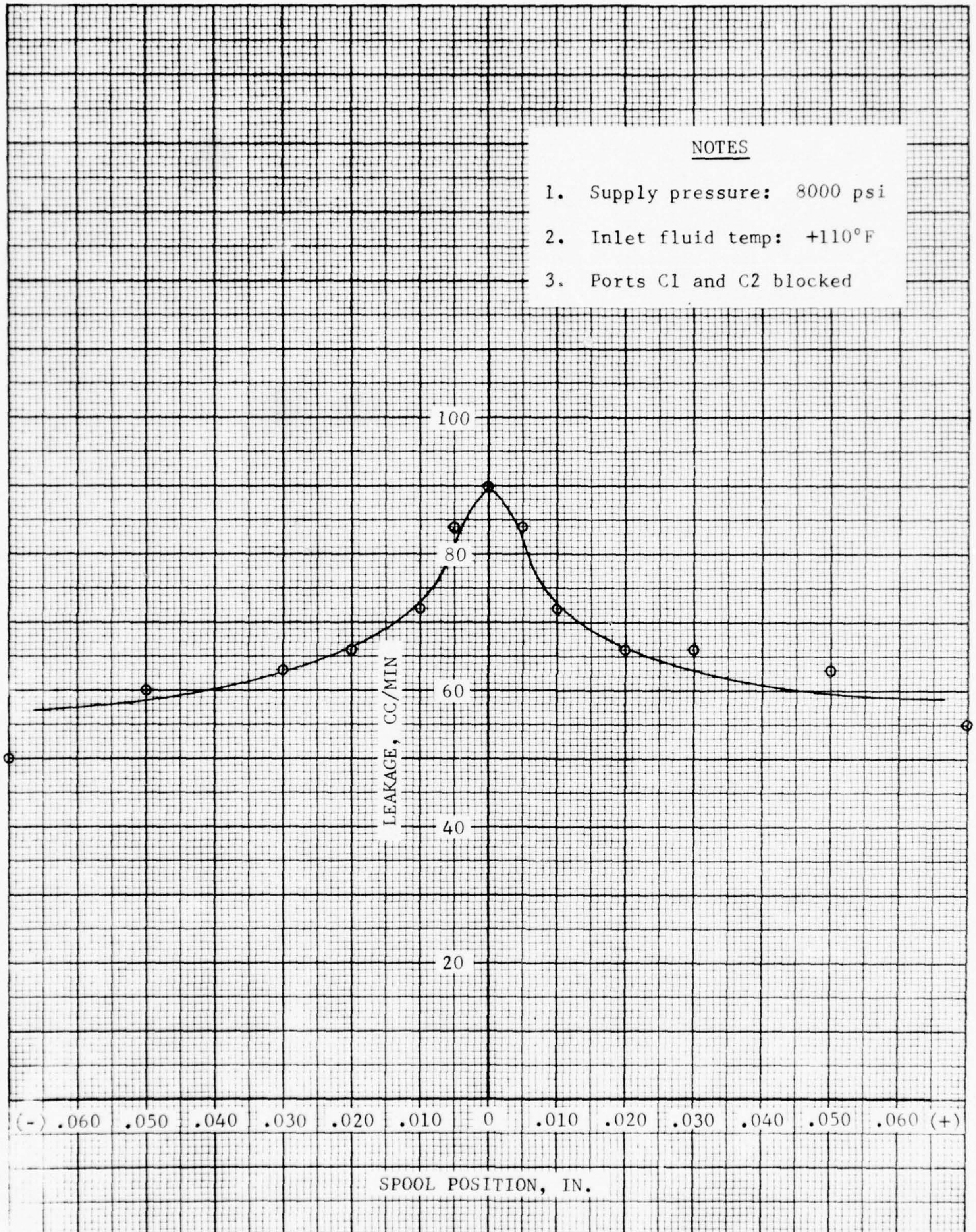


FIGURE 24 INTERNAL LEAKAGE

5.0 SYSTEM TESTS

5.1 DESCRIPTION

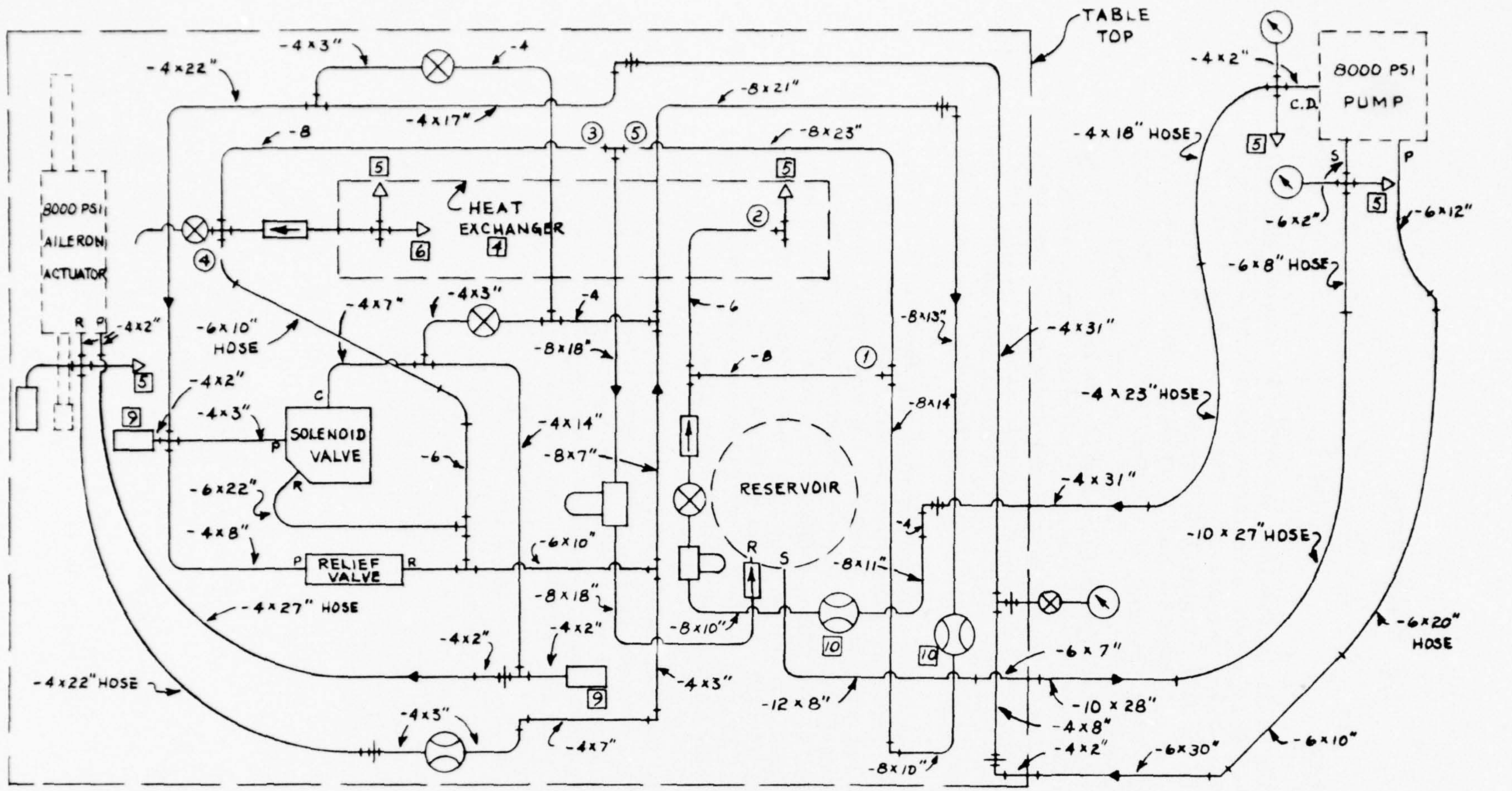
The laboratory setup built to simulate the 8000 psi lateral control system to be installed in a T-2C airplane is shown schematically on Figure 25 and pictorially on Figure 26. The system contains all of the hardware listed in Table I plus T-2C hydraulic system reservoir P/N 288-580600-11, an oil-to-water heat exchanger, and two 10 micron (nominal) filters. Other portions of the T-2C 3000 psi system (pump, elevator actuator, utility functions, etc.) were not included in the laboratory setup because the potential benefits derived from testing a complete system did not warrant the added expense. Temperature, pressure, and fluid flow instrumentation were installed at several locations in the setup; this equipment did not simulate flight test instrumentation, Section 2.2. Three needle valves were added to the system to permit conducting various tests on the pump, Section 3.0.

Line lengths and sizes to be used in the aircraft test installation were duplicated (as nearly as practical) in the laboratory setup. Tubing bends were not simulated, however all elbow type fittings to be used in the aircraft were put in the laboratory system. High pressure tubing was 21-6-9 CRES: tubing sizes were 1/4 X .025 and 3/8 X .038. All 8000 psi tubing connectors were standard MS flareless fittings; 8000 psi fittings to be used in the aircraft will be "Dynatube" series. Static seals were MS 28778 O-rings. The system contained 2.0 gallons of MIL-H-83282 hydraulic fluid.

Fluid flow rate in the pump case drain, system return, and actuator return lines was measured by turbine meters with readout on frequency counters. Static pressures were monitored with bourdon tube dial gages teed into the pump inlet, discharge, and case drain lines. Dynamic pressures were recorded on an oscillograph using strain gage type transducers plumbed in near the shutoff valve inlet port, inlet of the actuator pressure hose, and actuator return port. Fluid temperatures were sensed by thermocouples at six locations: pump inlet, discharge, and case drain, heat exchanger inlet and outlet, and aileron actuator return. Temperature readout was on a multi-channel, selector-button type indicator. Fluid temperature stabilization was achieved by means of a duration-adjusting type controller and an oil-to-water heat exchanger.

Return circuitry plumbing was designed to permit three different configurations to be used:

- System test with no heat exchanger
- System test with a heat exchanger in the pump case drain line
- Pump performance test with a heat exchanger in the system return line



NOTES

1. 5, 9, ETC. SEE FIGURES 13 AND 14
2. ALL 8000 PSI LINES: 1/4 x .025 21-6-9 CRES
   3/8 x .038 21-6-9 CRES
3. ALL RETURN LINES: 6061-T6 ALUM
4. LINE SIZES: -4 = 1/4 DIA.
   -6 = 3/8 DIA.
   -8 = 1/2 DIA.
5. UNION OR REDUCER
   90° ELBOW
   90° BULKHEAD ELBOW
   45° ELBOW
   TEE
   TUBING BEND
6. RETURN LINE CONFIGURATIONS:
   a) HEAT EXCHANGER IN RETURN LINE — CONNECT LINES AT 1, 2, 3
   b) HEAT EXCHANGER IN PUMP C.D. LINE — CONNECT LINES AT 2, 4, 5
   c) NO HEAT EXCH. — CONNECT LINES AT 5. CONNECT 2 WITH 4.

FIGURE 25 SCHEMATIC OF SIMULATED TEST SYSTEM INSTALLATION

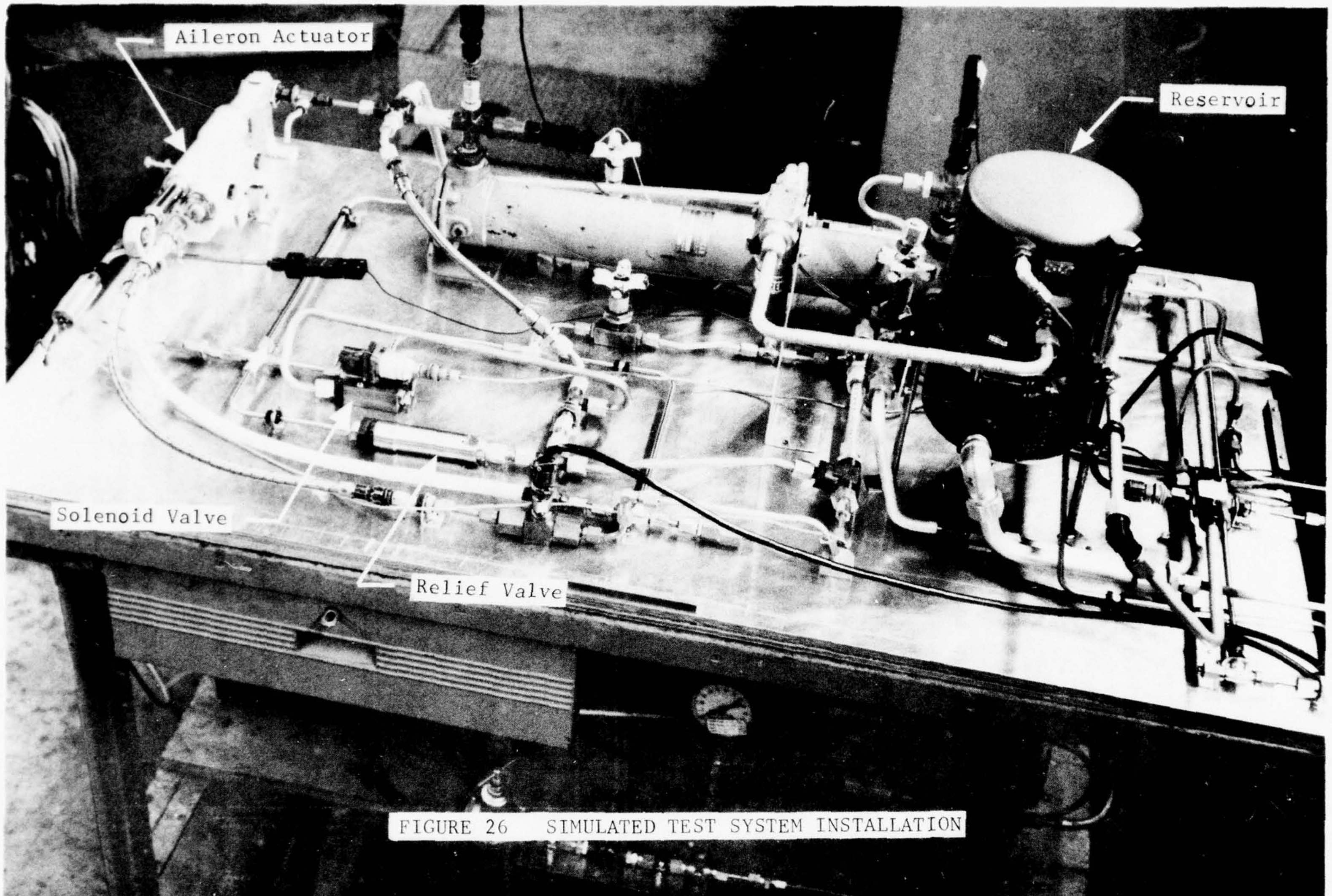


FIGURE 26 SIMULATED TEST SYSTEM INSTALLATION

System testing was conducted under two conditions: +110° and 180°F pump inlet fluid temperatures. The T-2C 3000 psi system normally operates with a +160°F fluid temperature in the reservoir. The +180°F temperature was based on a maximum allowable pump case drain temperature of +250°F in the 8000 psi test installation. The +110°F temperature was used for baseline data.

5.2 OPERATIONAL TESTS

5.2.1 System Stability

Plumbing details shown on Figure 4, i.e., line sizes fittings, valves, etc., will be included in the T-2C test system. This is not the original configuration designed, however. Original circuitry contained -4 size lines (only) in the 8000 psi portion of the system. This configuration was found, during preliminary laboratory testing, to produce resonance at pump speeds from 3200 to 6500 rpm. The resonance was caused by excitation of system fundamental and harmonic frequencies by pump ripple. Ripple was generally less than ±100 psi under stable conditions, however when resonance occurred, pressure oscillations up to ±500 psi at 800 to 1200 cpm were observed. Attenuation of these oscillations was achieved by adding fluid volume near the pump discharge port. This was done by changing -4 size lines (and hose) to -6, Figure 4.

Composite oscillograms of pump ripple and system resonance in the initial and final plumbing configurations are shown on Figures 27 and 28. This data was taken with the solenoid valve energized (8000 psi applied to the aileron actuator). Resonance was significantly greater at +180°F than at +110°F. This was attributed to less damping due to reduced fluid viscosity.

The amplitude and frequency of system resonance was found, during the laboratory investigation, to be sensitive not only to the amount of fluid volume added, but also to the location where the volume was placed. This suggests that although the laboratory system was made stable, some tuning may be required in the T-2C, since the test installation will not be identical to the laboratory setup.

5.2.2 Pressure Surges

Pressure fluctuations will occur as a result of (1) operation of the solenoid valve and (2) operation of the aileron actuator. Peak surges were measured using high response instrumentation, Figure 14. A summary of the surge data is presented in Table VI. The maximum surge observed in the pressure line was 8800 psi and occurred when the solenoid valve was energized. This is 110% of system pressure and is under the 120% maximum allowable, Reference 4. The peak surge in the return line was 370 psi; this occurred when the solenoid valve was de-energized. No significant surges were generated by operation of the aileron actuator.

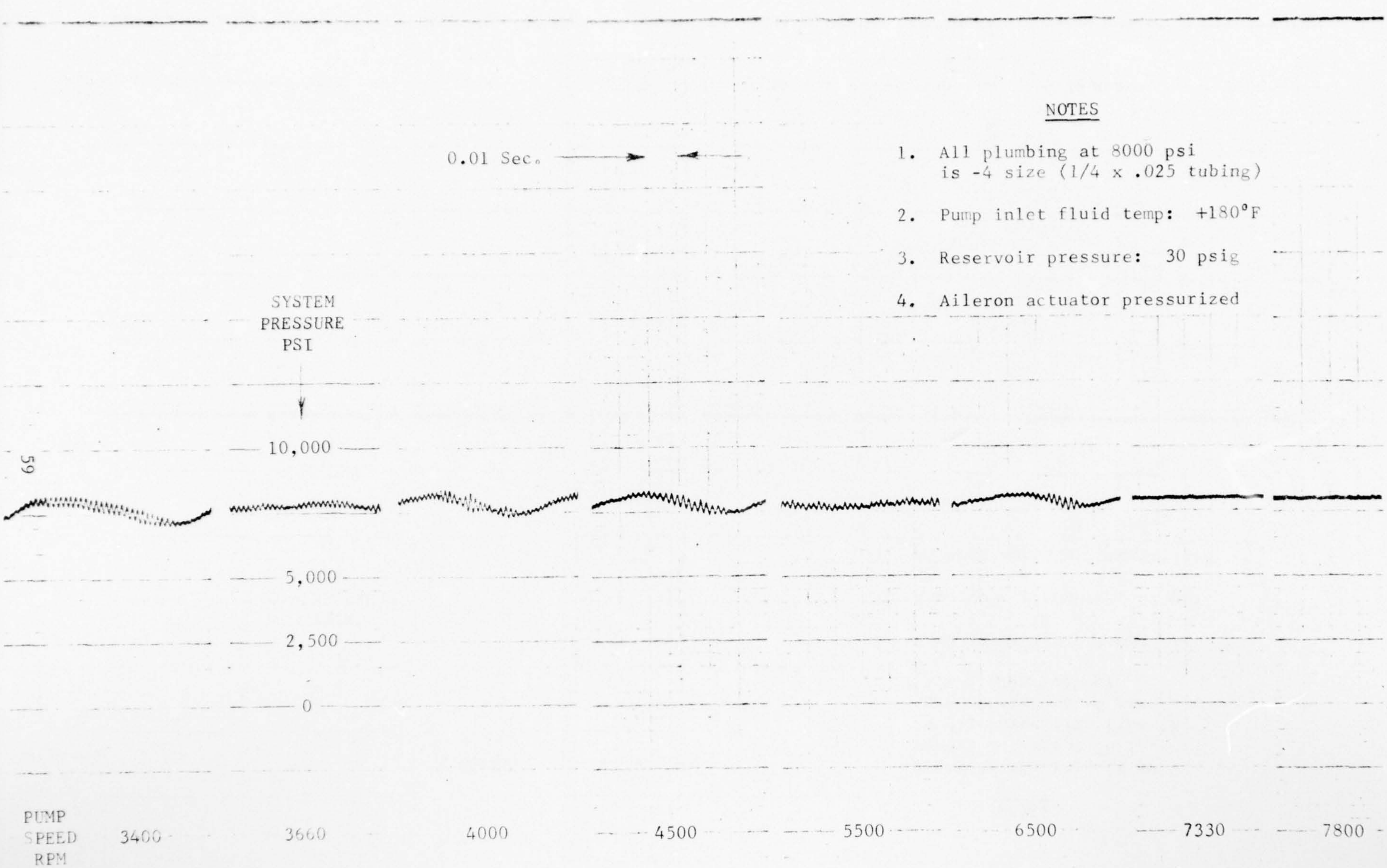


FIGURE 27 SYSTEM RESONANCE WITH INITIAL PLUMBING CONFIGURATION

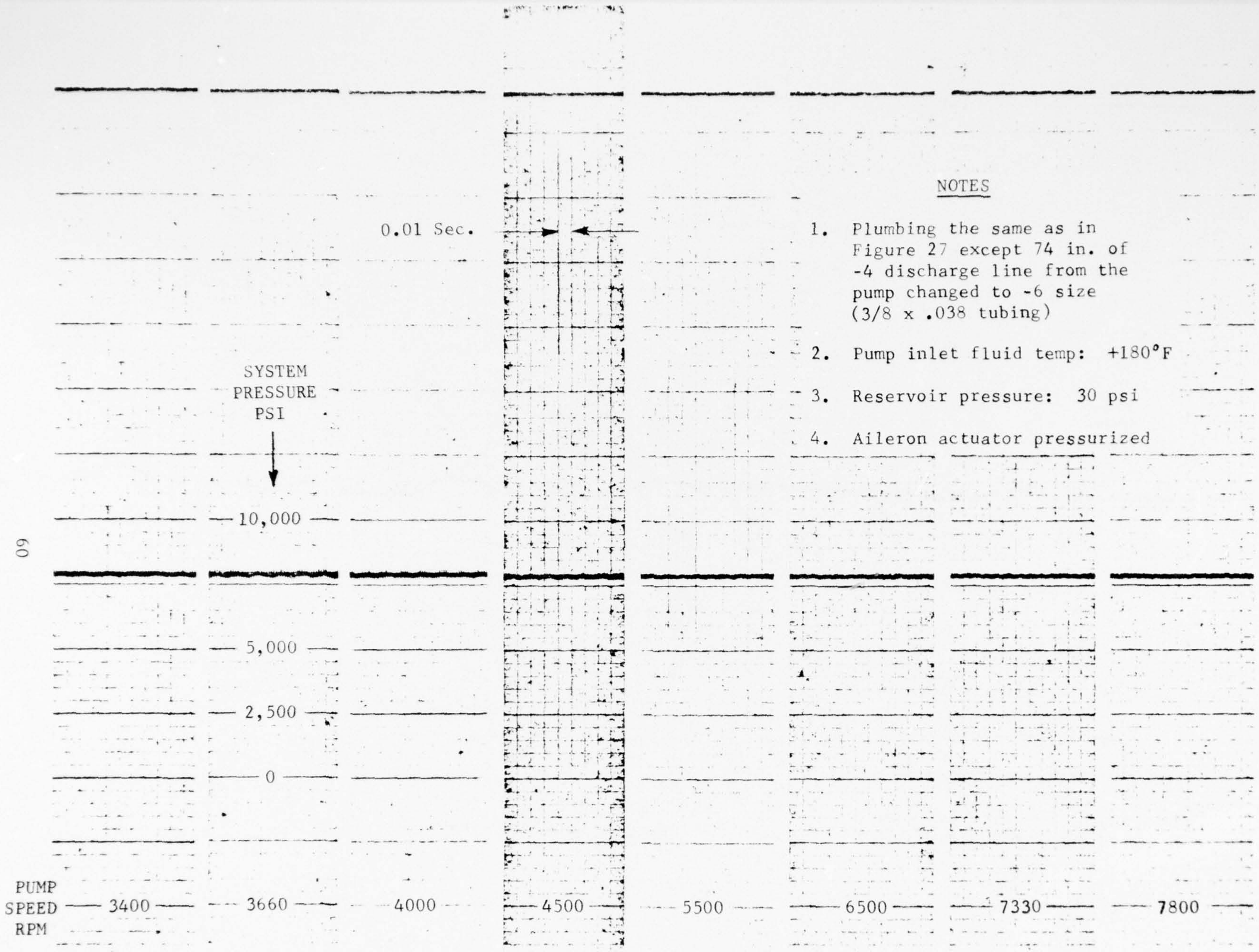


FIGURE 28 SYSTEM STABILITY IN FINAL PLUMBING CONFIGURATION

TABLE VI

DATA SUMMARY OF PRESSURE SURGE TESTS

PUMP INLET FLUID TEMP.	PUMP SPEED, RPM	CONDITION	P_{in} PSIG	P_{cyl} PSIG	P_r PSIG
+110°F	3600	1	8400	8400	30
		2	8000	0	230
		3	8000	8000	60
		4	8100	8100	60
	7330	1	8800	8800	30
		2	8000	0	180
		3	8100	8100	60
		4	8100	8100	70
+180°F	3660	1	8200	8200	40
		2	8000	0	370
		3	8100	8100	80
		4	8200	8200	80
	7330	1	8700	8700	40
		2	8000	0	370
		3	8100	8100	80
		4	8100	8100	80

NOTES

1. Pressure values are peak pressures.
2. P_{in} = Pressure upstream of solenoid value
   P_{cyl} = Pressure half way between sol. valve & aileron actuator
   P_r = Pressure at actuator return port.
3. Condition 1 = Solenoid valve energized
   2 = Solenoid valve de-energized
   3 = Actuator driven max rate stop-to-stop
   4 = Actuator driven hard oscillations at mid-stroke

5.2.3 System Temperatures

The thermal analysis presented in Section 2.3 indicated that a heat exchanger would be required in the T-2C test installation. Two tests were conducted to verify the analysis: (1) determination of the temperature/time function with no heat exchanger in the system; (2) determination of system temperatures when pump inlet fluid is stabilized at +180°F using a heat exchanger in the pump case drain line. Both tests were run with the plumbing exposed to quiet, room temperature air. The aircraft system will experience moving air; this will increase heat removal. The surface area and volume of T-2C 3000 psi circuits (not included in the laboratory setup) will also contribute to lowering stabilization temperatures. The laboratory tests thus produced conservative results.

Test data with no heat exchanger are presented below; pump speed was 3660 rpm; case drain pressure was 80 psig.

Elapsed	Temperatures, °F				Flows, GPM	
Time, Min.	Ambient Air	Pump Inlet	Case Drain	Actuator Return	Case Drain	Actuator Null
0	78	77	115	80	.34	.03
5	94	117	203	90	.51	.04
10	109	193	258	112	.69	.06

Since a heat exchanger was obviously required, no further testing was conducted without one. Steady state temperatures observed with a heat exchanger in the pump case drain line, inlet fluid temperature stabilized at 180°F, and 80 psig case drain pressure are listed below:

Pump	Temperatures, °F						Flows, gpm	
Speed, rpm	Ambient Air	Pump Inlet	Case Drain	Act. Ret.	H.Exch. Inlet	H.Exch Outlet	Case Drain	Act. Null
3660	98	180	250	167	247	193	.635	.088
7330	100	180	249	171	246	189	.735	.093

Heat removed by the heat exchanger was calculated as follows:

$$HR = WQC_p\Delta T$$

where, HR = heat removed, BTU/min

W = fluid density, lb/in^3

Q = flow rate, in^3/min

C_p = specific heat, BTU/lb/°F

ΔT = fluid temperature drop across heat exchanger, °F

now, W = $.0284\ lb/in^3$ @ +219°F *

C_p = .555 BTU/lb/°F @ +219°F *

*Data from Reference 10.

thus,

@ 3660 rpm HR = $.0284 \times 231 \times .635 \times .555 \times 54$

= 125 BTU/min

@ 7330 rpm HR = $.0284 \times 231 \times .735 \times .55 \times 57$

= 153 BTU/min

As stated previously, the laboratory test results are conservative due to (1) still air environment, and (2) 3000 psi circuits not included in the laboratory setup. The results at 3660 rpm (above) agree well, however, with the analysis in Section 2.3 which estimated that 122 BTU/min should be removed during ground idle. The results at 7330 rpm cannot be compared with the thermal analysis, since the laboratory environment was not representative of cruise conditions at 15,000 ft.

5.3 ENDURANCE EVALUATION

5.3.1 Pump

5.3.1.1 Abex Tests

Pump M/N APIV-106 had a bronze barrel and 0.99 CIPR displacement during the first step in its development. Tests conducted on this configuration were (1) break-in run, (2) calibration, (3) minimum inlet pressure, (4) pressure ripple, and (5) transient response, Reference 8. Total operating time accumulated during these tests was 18.5 hours.

A steel barrel containing bronze sleeves was then fabricated and installed in the pump, and displacement was increased to 0.124 CIPR. Total operating time on this configuration was 9.5 hours.

Pump operating parameters at Abex ranged from +120 to +220°F inlet fluid temperatures, and from 2400 to 10,000 rpm. Total running time on the pump was 18.5 + 9.5 = 28 hours.

5.3.1.2 CAD Tests

Pump operating time accumulated at CAD is summarized below:

Section		Total Time, Hours
3.2	Pump Performance	4.6
5.2.1	System Stability Investigation	5.5
5.2.2	System Pressure Surge Tests	0.5
5.2.3	System Temperature Tests	2.9
5.3.2	Simulated Flight Tests	10.5
	Total	24.0

CAD time plus Abex time totaled 52 hours. Pump performance was satisfactory at the conclusion of laboratory testing, Section 5.3.2.

5.3.2 Simulated Flights

Performance of the 8000 psi lateral control system will be evaluated in a T-2C airplane during the next phase of the LHS program. A total of approximately 10 flight hours will be accumulated. The following is a cycling schedule established to profile a typical flight.

Flight Phase	Duration, min	Pump Speed, rpm	Actuator Operation
Ground Checkout & Taxi Out	15	3660	Periodic
Takeoff	7.5	7800	Continuous
Cruise	45	7330	Periodic
Full Power Flight	7.5	7800	Periodic
Landing & Taxi In	15	3660	Periodic
Total	90		

The above schedule was used in the laboratory simulated flights. All tests were run with minimum reservoir pressurization (27 psia) and 80 psig maximum in the pump case. Because the pump was oversized, Section 3.1, a heat exchanger was used to maintain fluid temperatures below +180°F at the pump inlet. System temperatures stabilized after the "takeoff" phase of a "flight." Typical values were:

Pump inlet fluid	+176°F
Pump case drain fluid	+255°F
Actuator return fluid	+156°F
Ambient (still air near reservoir)	+ 94°F

System performance was evaluated from the following criteria:

(1) Pump case drain flow

(2) Actuator null flow

(3) External leakage (pump and actuator)

(4) Internal leakage (solenoid valve and relief valve)

(5) Pump case drain filter bowl debris

(6) Fluid contamination level

Performance data taken before and after seven 1½ hour simulated flights are summarized in Table VII.

TABLE VII

SIMULATED FLIGHT PERFORMANCE SUMMARY

	BEFORE TEST	AFTER 10.5 HOURS
Pump Case drain flow, gpm (at +255°F and 7330 rpm)	0.65	0.67
Actuator null flow, cc/min (at +156°F, includes rod seals)	152	241
External leakage, pump	None	None
External leakage, actuator	None	Trace (at rod seal)
Internal leakage, solenoid valve, drops per minute at room temp		
P → R	6	5
C → R	8	8
Internal leakage, relief valve, at room temperature	2 drops/min	60 cc/min
Case drain filter bowl debris (visual examination)	Clean	Normal quantity of wear particles present

Fluid contamination:

PARTICLE SIZE, MICRONS

	5-10	10-25	25-50	50-100	100+
*Class V	87,000	21,400	3130	430	41
Before	55,702	5,892	50	1	0
After	10,803	309	12	1	0

*NAVAIR 01-1A-17

Pump performance was satisfactory throughout the test. System pressure fluctuations and surges were nominal as anticipated. The pump had no external leakage. A normal quantity of metallic wear particles were observed in the case drain filter bowl following the test.

Actuator internal leakage increased from 152 cc/min before to 241 cc/min after the test. These values represent control valve null leakage plus the leakage of two metallic rod seals. The actuator was disassembled for examination. All static seals were in satisfactory condition. Null leakage of the control valve was checked and found to be unchanged from that shown on Figure 24. The increase in actuator internal leakage was therefore presumed to be at the rod seals. These seals originally had a 0.003 in. interference fit on the rod. This fit produced excessive friction during preliminary testing and was subsequently reduced to 0.001 in. interference. The fit modification reduced seal friction considerably, but increased seal leakage. Seal wear-in probably caused the leakage change observed during the simulated flights. The 89 cc/min higher leakage rate had no detrimental affect on system performance. A trace of external leakage was observed at the rod seal; this was considered normal and desirable.

Solenoid valve performance was satisfactory. Internal leakage was not changed by the test.

Relief valve internal leakage increased from 2 drops/min to 60 cc/min. The valve was disassembled after the test and examined. No evidence of unusual wear was found; all static seals were in satisfactory condition. The valve was re-assembled and re-tested. Leakage at 8000 psi was a normal 5 drops/min. The cause of the leakage anomaly was not determined.

Fluid contamination level decreased substantially during the test. This was evidence of (1) the filters performing satisfactorily, (2) little contamination being generated, and (3) good fluid lubricity.

6.0 DISCUSSION

Work on the LHS concept was begun with a theoretical analysis of factors associated with the development of very high pressure fluid power systems, Reference 1. Subsequent studies showed that significant weight savings could be gained by operating at 8000 psi instead of the conventional 3000 psi level. Work progressed in logical phases from theoretical considerations to evaluation of experimental hardware to endurance testing. Information in this report deals with preparations for flight testing an LHS lateral control system in a T-2C airplane. A brief review covering the state-of-the-art of lightweight hydraulic system development is appropriate at this point. Areas to be discussed include (1) analysis and design and (2) hardware development.

6.1 ANALYSIS AND DESIGN

Several design parameters are affected by operating pressure level -- in particular, fluid viscosity, actuator stiffness, pressure surges, and heat generation. The degree to which 8000 psi (versus 3000 psi) affects these parameters and thus system performance is a primary concern. Theoretical analyses were made in 1966, Reference 1. Since then, many hours of laboratory testing and investigation have provided new insights into the effect of high pressure on these parameters.

6.1.1 Fluid Viscosity

Viscosity is directly related to tubing pressure losses and component internal leakage rates. Temperature has a marked influence on fluid viscosity; pressure has a less pronounced effect (the kinematic viscosity of MIL-H-83282 is about 1½ times greater at 8000 psi than at 3000 psi). Classic fluid flow theory for pressure drop, orifice flow, capillary flow, etc., is applicable at 8000 psi. Thus, procedures used at 3000 psi to determine line losses, orifice sizes, and valve dimensions are also valid at 8000 psi.

Laminar flow losses are less at 8000 psi than at 3000 psi for a given tube size and transmitted power level. If tube size is reduced in accordance with the lower flow requirements at 8000 psi, then laminar flow losses at 8000 psi are slightly higher (percentage-wise) than those at 3000 psi. Turbulent losses are not affected by operating pressure level for a given tube size and flow rate, Reference 4. If tube size is reduced in accordance with the lower flows at 8000 psi, then turbulent flow losses are less (percentage-wise) than those at 3000 psi. Line pressure drops are normally established by the minimum operating temperature since losses are less at any higher temperature. Heat generated by losses in cold 8000 psi tubing will rapidly increase fluid temperatures and reduce these losses. The pressure effect on viscosity is not a factor in return lines; lower flow rates will therefore permit the use of smaller diameter return tubing in 8000 psi systems.

The effect of high operating pressure on line losses may have been somewhat overstated by the Reference 1 study. In real world applications, normal rapid warmup of an operating system should dimish the importance of this variable.

Viscosity increase due to pressure has a beneficial effect in reducing leakage through small clearances. This means that 8000 psi components do not require special fits and tolerances to maintain acceptable internal leakage rates. This has been demonstrated by the pumps, actuators, and valves built for the LHS program; internal leakage observed was nominal using conventional clearances and tolerances.

6.1.2 Actuator Stiffness

Weight savings produced by operating at 8000 psi instead of 3000 psi begins with smaller net areas on actuator pistons. Lower flow demand because of less displaced fluid results in a general decrease in the size of supply lines, pumps, reservoirs, etc. Use of a smaller piston area, however, reduces actuator physical stiffness which in turn lowers system resonant frequency. The following paragraphs will attempt to show that this reduction in actuator stiffness will not significantly degrade system performance.

Mechanical elements which contribute to physical stiffness include bearings, piston/rod, cylinder, and actuator end pieces; hydraulic elements are the fluid and seals. Based on practical experience, it has been found that actuator physical stiffness is approximately equal to the stiffness due to fluid bulk modulus. Thus,

$$K_f = \frac{4 \beta_f A \eta}{S}$$

Where,

K_f = Stiffness due to fluid compressibility (actuator piston at mid-stroke)

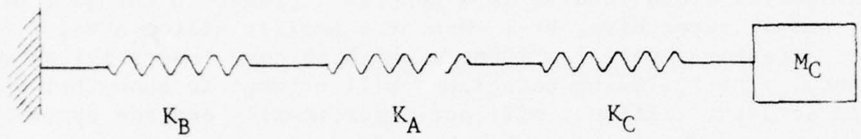
β_f = Bulk modulus of fluid (taken at one-half system pressure, approximately 15% higher at 4000 psi than at 1500 psi)

A = Piston net area

S = Piston Stroke (total)

η = Ratio of fluid volume swept by the piston to the total fluid volume contained between the piston and control valve, typically between 0.85 and 0.95 for an integrally mounted valve

In a practical hardware installation, the actuator backup structure, the actuator, and the control surface are three significant springs, as shown below. Control surface inertia is the only significant mass. The three springs are in series, anchored to the aircraft at one end and supporting the mass at the other. If the spring rates of each are assumed equal, then the actuator is twice as stiff as the combined structure/surface spring. Normally, however, the actuator is the stiffest spring in the system.



Where,

K_B = Backup structure spring rate

K_A = Actuator spring rate

K_C = Control surface spring rate

M_C = Control surface mass

Now,

$$\frac{1}{K_T} = \frac{1}{K_B} + \frac{1}{K_A} + \frac{1}{K_C}$$

Where, K_T = Net total spring rate

Now let

$$\frac{1}{K_{BC}} = \frac{1}{K_B} + \frac{1}{K_C}$$

If,

$$K_A = K_B = K_C$$

Then,

$$K_A = 2\, K_{B_C}$$

System natural frequency is established by net total stiffness and varies, not directly, but with the square root of the stiffness.

$$\omega_n = \sqrt{\frac{K_T}{M_e}}$$

Where, ω_n = System undamped natural frequency

K_T = System net spring rate

M_e = Effective mass

Physical stiffness is the composite effect of mechanical and hydraulic compliant elements between the actuator mounting points. Actuator functional stiffness is due to closed loop servo action, and is related to loop gains and control valve performance characteristics. In a conventional position feedback system, functional static stiffness is higher than physical stiffness.

Frequency response tests of similar sized "muscle" actuators operating at different pressure levels were conducted in a rigid mass load fixture, Reference 3. Damped natural frequencies observed were:

Actuator	Operating Pressure	Resonant Frequency
CAD P/N 247-58716	3000 psi	27 Hz
CAD P/N 4212-01	6000 psi	24 Hz
CAD P/N 4212-01	9000 psi	22 Hz

Performance characteristics of the 6000 and 9000 psi actuators were very similar to the 3000 psi actuator for large amplitude, manual-type inputs. For small amplitude inputs, such as those encountered with automatic control, response capabilities of the 6000 and 9000 psi actuators were satisfactory to 10 Hz. The reduction in resonant frequency noted above is therefore not considered critical.

The various points made in this discussion suggest that undue attention should not be focused on actuator stiffness. Although physical stiffness is fundamentally reduced by going to higher operating pressures, this is partially offset by an increase in fluid bulk modulus. The net reduction should not degrade system performance significantly in conventional applications. Furthermore, control techniques are available to offset lag effects caused by reduced stiffness.

6.1.3 Pressure Surges

Pressure surges are normal in aircraft hydraulic systems and are an important design parameter because of their affect on the fatigue and functional characteristics of system components. Surges result primarily from:

(1) Sudden stopping of high velocity fluid

(2) Sudden porting of high pressure fluid into a chamber filled with low pressure fluid

(3) Bottoming of an actuator piston

(4) External energy derived from load inertia

When the flow of a mass of fluid is suddenly decelerated by a rapidly closing valve, water hammer results; this is usually the most severe pressure transient encountered in hydraulic circuits. Assuming instantaneous valve closure, this surge may be calculated by,

$$\Delta P = V\sqrt{\rho \beta_e}$$

Where,

ΔP = Maximum pressure rise above system pressure

V = Fluid velocity

ρ = Fluid mass density

β_e = Effective bulk modulus (fluid compressibility + tube elasticity)

Pump response time is also a factor causing surges. Operation of the delivery control mechanism normally occurs in 0.050 sec. or less. Thus, when a valve closes, the pump momentarily continues to discharge fluid until the control mechanism adjusts to the new flow demand; this can result in a pressure overshoot.

Surges in 8000 psi systems are less, percentage-wise, than in 3000 psi systems because of:

(1) Better damping at 8000 psi due to increased fluid viscosity

(2) The minor effect of operating pressure level on water hammer magnitude

(3) Faster pump response at 8000 psi

Typical peak surges observed in 3000 and 8000 psi systems are compared below:

System	Peak Pressure	Overshoot
3000 psi	3900 psi	130%
8000 psi	9100 psi	114%

The maximum allowable surge in 3000 psi systems is 135%, reference MIL-H-5440E. The maximum allowable surge in 8000 psi systems is 120%, Reference 4. The validity of the 120% design value has been confirmed many times by laboratory testing.

6.1.4 Heat Generation

Hydraulic systems generate heat because it is impossible to convert all input power into useful work. Thus, all hydraulic systems normally operate at temperatures above ambient. Temperature stabilization is reached when the heat loss rate equals the generation rate. Hydraulic fluid temperatures must be maintained below stated maximums to prevent thermal breakdown of the fluid and seals. For Type II systems this temperature is +275°F; for Type III systems it is +390°F. If heat dissipation through conduction, radiation, and convection is not sufficient to maintain reasonable fluid temperatures, then a heat exchanger is required. A hydraulic system must be designed so that a heat balance is achieved at a satisfactory operating temperature.

The principal sources of heat generation in hydraulic systems are:

(1) Pump and valve internal leakage

(2) Orifices and valves used to throttle and control flow (These devices are inherent heat generators.)

(3) Resistive pressure drops in lines, fittings, and porting passageways

The principal means of heat dissipation are:

(1) Conduction from hydraulic system components through attachments into aircraft structure

(2) Convection aided by air flow around system components

(3) Radiation from system components

In aircraft where variable delivery pumps drive servo actuator systems, nearly all input power is eventually dissipated as heat (under steady state operating conditions). When a system has no work output, then equilibrium is attained when system temperature is high enough above ambient to cause a heat transfer rate equal to the energy input rate at the pump. Thus,

$$W_{in} = Q_{loss} = U A (T_{sys} - T_{air})$$

Where,

W_{in} = Work input

Q_{loss} = Heat loss

U = Overall coefficient of heat transfer

A = Surface area of system

T_{sys} = System temperature (average)

T_{air} = Ambient air temperature

Since operating temperatures are related directly to the surface area of a hydraulic system, cooling requirements will be somewhat greater at 8000 psi due to the inherent compactness of the system (assuming 3000 psi and 8000 psi pump efficiencies are the same). Therefore, 8000 psi systems must be designed to operate at slightly higher ΔT allowables.

6.2 HARDWARE DEVELOPMENT

Several companies provided notable cooperation during development of components used in the LHS program. This cooperation has been a major factor in the success of the program.

COMPANY	HARDWARE
Aerospace Division of Abex Corporation	Pumps
Sterer Engineering and Manufacturing Co.	Solenoid Valves
PneuDraulics, Inc.	Relief Valve
The Lee Company	Restrictors
W.S. Shamban and Co.	Seals
Greene, Tweed & Company	Seals
Cook-Airtomic Division of Dover Corporation	Seals
Rosán, Inc.	Fittings
Resistoflex Corporation	Fittings & Hoses
Titeflex Division of Atlas Corp.	Hoses

Hardware items to be discussed in this section include the pump, fluid, seals, actuators, and tubing.

6.2.1 Pump

Two variable delivery pumps have been developed for use in the LHS program:

Abex M/N AP6V-57, rated for 14 gpm at 7850 psi and 4000 rpm

Abex M/N APIV-106, rated for 3 gpm at 7850 psi and 7330 rpm

Both pumps functioned satisfactorily and had overall efficiency levels comparable to 3000 psi units. The pump designs were conventional and no significant problems were encountered in their development. Pump endurance has not yet been fully explored. This area is not expected to produce any serious difficulties since performance thus far has been very similar to that of 3000 psi units.

Both 8000 psi pumps were developed from existing hardware. No effort was made to optimize design or to minimize weight; cost was a primary concern. A pump designed from inception to operate at 8000 psi would more fully realize the weight and space saving benefits of the LHS concept.

6.2.2 Fluid

Shear stability is particularly important in a fluid required to operate at high pressure levels. MIL-H-83282 has exhibited excellent shear stability in testing conducted thus far. This fluid also provides good lubricity as evidenced by the low wear observed during hardware endurance tests. It is anticipated that further testing should establish MIL-H-83282 as a satisfactory fluid for 8000 psi hydraulic systems.

6.2.3 Seals

Dynamic seals were expected to be a critical area in hardware development. As a result of judicious design, this has not been the case. The use of metallic piston seals proved to be a very satisfactory approach. No difficulties have arisen with the currently used two-stage metallic/elastomeric rod seal, however further testing is needed (1) to evaluate the potential of single stage seals, and (2) to evaluate non-metallic two-stage rod seals.

Conventional elastomeric O-rings have been satisfactory as static seals. Standard teflon backup rings were employed successfully in diametral applications. In boss type seals, it was found that careful attention must be exercised to eliminate any clearance through which the O-ring could extrude.

6.2.4 Actuators

Four (4) 8000 psi servo actuators have been designed and built by CAD, Table VIII. Two of the units were for the Navy sponsored Advanced Flight Control Actuation System (AFCAS) development program, Reference 9 and 11.

A principal objective has been to show that 8000 psi actuators can be built to meet specific performance requirements using conventional design practices and manufacturing procedures. Although component weight reduction is the fundamental reason for the LHS program, no attempt was made to optimize the actuators; cost was a primary consideration.

Each of the 8000 psi actuators has an integrally mounted, single stage, spool/sleeve type flow control valve. Actuator cylinder walls were sized for burst; rods were designed for column buckling. Two-stage rod seals and three piece metallic piston seals were used in all units. Each actuator configuration was defined by (1) program objectives, (2) maximum hinge moment, (3) stroke length, (4) maximum piston velocity, (5) available space, and (6) dynamic performance requirements.

Detail design procedures were similar to those employed for 3000 psi units. Piston areas were sized so that full load could be carried at rated speed with a differential pressure across the piston of 2/3 system pressure. Fluid volume between the control valve and actuator was kept small to maximize actuator stiffness. Leakage across the first stage rod seal provided lubrication; this leakage was minimal and did not represent a significant power drain. Provisions for porting rod seal leakage to return were simple and straight-forward.

Control valve performance was expected to be a problem at 8000 psi because of (1) high internal leakage and (2) metering edge erosion. These problems did not develop. Null power loss has been nominal -- generally less than 0.12 hp. Valve erosion has been negligible. The use of MIL-H-83282 fluid may have contributed to valve performance through its excellent shear stability and lubricity characteristics.

Design of the control valve orifices was conventional. Spool/sleeve clearances and tolerances were the same as those used for 3000 psi units. Valve overlap (0.002 in.) was employed to insure stability at null and minimize internal leakage.

Operation of the 8000 psi actuators was, from outward appearances, identical to similar sized 3000 psi units. Spool flow forces were low, output piston motion was easily controlled, resolution was good, and dynamic response was satisfactory. Actuator endurance is one area yet to be examined. Endurance testing will provide important information on seal performance, control valve wear, and fatigue of structural elements.

TABLE VIII

8000 PSI ACTUATORS DESIGNED AND BUILT AT CAD

CAD P/N	TYPE	INPUT	MID-STROKE LENGTH, IN	PISTON STROKE, IN	MAX. OUTPUT FORCE
4212-01	For lab tests, dual system tandem, balanced piston	Manual	46.2	8.2	26,000 lb/chamber
*4248-01	For lab tests, dual system tandem, partially balanced, modular construction	Electrical (torque motor driven valve)	38.3	8.2	46,000 lb extend 36,000 lb retract
4257-01	For T-2C aileron, single system, balanced piston	Manual	15.2	3.0	1870 lb
*4262-01	For T-2C rudder, single system, balanced piston, modular construction	Electrical (torque motor driven valve)	16.6	3.5	1870 lb

*Built for AFCAS Program

6.2.5 Tubing

The following discussion is presented to strengthen the assertion that a tubing burst factor of 3 for 8000 psi systems is equivalent, safety-wise, to the burst factor of 4 used in 3000 psi systems.

Special size tubing was fabricated for the LHS program. Use of this tubing permits the transmission of nearly twice the horsepower per pound of tubing at 8000 psi than at 3000 psi (for a given fluid velocity and tube material). Tubing wall thickness was determined using a burst pressure of 24,000 psi -- 12,000 psi is used for 3000 psi systems. The burst factor used for 8000 psi tubing was based principally on the fact that pressure surges encountered in 8000 psi systems are smaller, percentage-wise, than in 3000 psi systems. Surprisingly, the burst factor of 3 provides nearly twice the pressure safety margin at 8000 psi than the factor of 4 does at 3000 psi, as shown below.

Operating Pressure, psi	Max. Allowable Press. Surge		Burst Pressure, psi	Pressure Safety Margin, psi
	Percent	psi		
3000	135	4050	12,000	7,950
8000	120	9600	24,000	14,400

Comparison of tube stresses in 1/4 X .020 304 CRES typically used in 3000 psi systems and 1/4 X .025 21-6-9 CRES proposed for 8000 psi systems are compared below:

Operating Press., psi	Tube Size	CRES Mat'l.	Mat'l. Ult. Strength, psi	Hoop Stress at Operating Press., psi
3000	1/4 X .020	304	105,000	17,250
8000	1/4 X .025	21-6-9	142,000	36,000

Fatigue properties of 1/4 X .020 304 CRES and 1/4 X .025 21-6-9 CRES have been evaluated by Resistoflex using the rotary flexure method, Appendix A. Tubing life appears to be infinite -- more than 10^7 cycles -- when the combined stress level (bending + axial stress) is below 48,000 psi. This is considered a satisfactory design maximum for most applications.

The human element in fabricating and installing tube assemblies can sometimes lessen the integrity of the tubing. Tolerances on tube diameters and material imperfections can reduce tubing strength. The burst factor used for 3000 psi tubing includes these influences and results in a thicker wall than would be required by pressure alone. Tubing for 8000 psi systems has thicker walls and thus inherently has the extra ruggedness required for "handling". This aspect permits a more efficient design, weight-wise.

7.0 RECOMMENDATIONS

Detail design of an experimental 8000 psi aileron control system has been completed, and laboratory testing of the system produced satisfactory results. The next step is to evaluate flight performance. The contractor therefore recommends the following tasks be performed in Phase IX of the LHS development program.

PHASE IX   <u>FLIGHT VERIFICATION OF LHS CONCEPT</u>

TASK I   <u>SYSTEM INSTALLATION</u>

1. Install experimental 8000 psi hydraulic system in T-2C, BuNo. 152382.
2. Install and calibrate necessary instrumentation

TASK II   <u>GROUND CHECKOUT</u>

1. Conduct preflight ground testing to assure satisfactory operation.
2. Establish flight test plan.
3. Obtain Navy release for flight

TASK III   <u>FLIGHT TEST</u>

1. Conduct flight evaluation program of 8000 psi aileron control system.
2. Determine system performance through both instrumentation and pilot observations.

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LIST OF ABBREVIATIONS

AFCAS	Advanced Flight Control Actuation System
BTU/min	British Thermal Units per minute
CAD	Columbus Aircraft Division
CBW	Control-by-wire
cc/min	cubic centimeters per minute
CD	case drain
CIPR	cubic inches per revolution
cpm	cycles per minute
CRES	corrosion resistant
C1	cylinder port No. 1
EGT	exhaust gas temperature
°F	degrees Fahrenheit
F.S.	fuselage station
ft/sec	feet per second
GFE	government furnished equipment
gpm	gallons per minute
hp	horsepower
hz	Hertz (cycles per second)
ICAO	International Civil Aviation Organization
in.	inch
in^2	square inches
in/sec	inches per second
lb	pound

L.H.	left hand
LHS	Lightweight Hydraulic System
L/R	left and right
max.	maximum
M/N	model number
NADC	Naval Air Development Center
OAT	outside air temperature
O.D.	outside diameter
P	pressure
P/N	part number
psia	pounds per square inch absolute pressure
psi	pounds per square inch (stress or pressure)
Δp	differential pressure
R	return
R.H.	right hand
rpm	revolutions per minute
S	suction
sec	second (time)
S.O.	shut-off
V/STOL	Vertical/Short Take-Off and Landing
W.L.	water line

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APPENDIX A

21-6-9 TUBING FLEXURE TESTS

(RESISTOFLEX DATA)

TUBE ASSEMBLY FLEXURE DATA

Resistoflex Corporation, Roseland, New Jersey

TUBE SIZE: 1/4 x .025 TUBE MATERIAL: 21-6-9 DATE OF TEST: October 1975

TYPE OF FLEXURE: Rotary TUBE SOURCE: Rockwell Columbus FITTING: MR54040T-04 REV: F

DESIGN BENDING STRESS: 20,000 PSI FITTING ATTACHMENT METHOD: Mechanically Swaged

AXIAL STRESS DUE TO FLUID: 14,192 PSI FLUID PRESSURE: 8000 PSIG CYCLE RATE 3000 (CPM)

COMMENTS: Proof Test. 12000 psig. Finished swage .217 - Room Temperature Burst Test using minimum swage conditions 30,000 psig plus (machine limit is 30,000 psig) No Failure

Specimen S/N	Installation Torque (Ft-Lbs)	Temp. (°F)	Dynamic Deflection (in.)	Dyn. Bending Stress (psi)	Combined Stress (psi)	Cycles 10^6	Comments
MA05	14 (Max.)	80	.057	19,500	33,692	10.28	No Failure
MA08	14 (Max.)	80	.082	26,250	40,442	10.26	No Failure
MA10	10 (Min.)	80	.109	34,500	48,692	4.56	Tube Failure .50 in. from Fitting
MA06	14 (Max.)	80	.112	36,000	50,192	1.65	Tube Failure (Swage Zone)
MA07	14 (Max.)	80	.124	40,500	54,692	.47	Tube Failure (Swage Zone)
MA11	10 (Min.)	80	.125	42,000	56,192	.68	Tube sheared at Fitting

NOTE:

1. Assemblies MA05 and MA08 subjected to 12,000 psi Proof test following completion of 10×10^6 flexure cycles.
2. Swage Zone means the portion of the tubing inside the fitting which is deformed into tube receptacle grooves.

TUBE ASSEMBLY FLEXURE DATA

Resistoflex Corporation, Roseland, New Jersey

TUBE SIZE: 3/8 x .038 TUBE MATERIAL: 21-6-9 DATE OF TEST: February 1976

TYPE OF FLEXURE: Rotary TUBE SOURCE: Rockwell Columbus FITTING: MR54040T-06 REV: F

DESIGN BENDING STRESS: 20,000 PSI FITTING ATTACHMENT METHOD: Mechanically Swaged

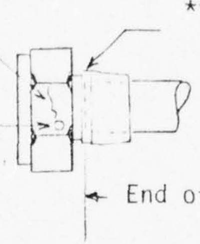
AXIAL STRESS DUE TO FLUID: 13,964 PSI FLUID PRESSURE: 8000 PSIG CYCLE RATE 3000 (CPM)

COMMENTS: All specimens proof tested at 12,000 psig prior to Rotary. Finished swage .321. Specimen MA71 was subjected to 12,000 psig proof test after flexure. Also, subjected to 30,000 psig Burst test - no failure after two applications of 30,000 psig.

Specimen S/N	Installation Torque (Ft-Lbs)	Temp. (°F)	Dynamic Deflection (in.)	Dyn. Bending Stress (psi)	Combined Stress (psi)	Cycles 10^6	Comments
MA71	15 (Min)	80	.220	30,000	43,964	10.260	No Failure
MA72	25 (Max)	80	.241	33,000	46,964	4.731	Fracture in Nut *
MA75	15 (Min)	80	.250	34,500	48,464	.608	Leak at seal surface #
MA76	15 (Min)	80	.249	34,500	48,464	10.260	No Failure
MA73	15 (Min)	80	.260	36,000	49,964	.494	Fracture in Shoulder**
MA74	25 (Max)	80	.291	40,500	54,464	.209	Fracture in Shoulder**

Damage occurred prior to test

*Fracture across flat of hex starting at Swivel Wire Hole



** Circumferential fracture in Shoulder in line with end of Tube

Swivel Wire Hole

End of Tube

AD-A032 677 ROCKWELL INTERNATIONAL COLUMBUS OHIO COLUMBUS AIRCRA--ETC F/G 1/3
DESIGN AND TEST OF AN LHS LATERAL CONTROL SYSTEM FOR A T-2C AIR--ETC(U)
MAY 76 J N DEMARCHI, R K HANING N62269-75-C-0422

UNCLASSIFIED NR76H-14 NADC-76166-30 NL

2 OF 2
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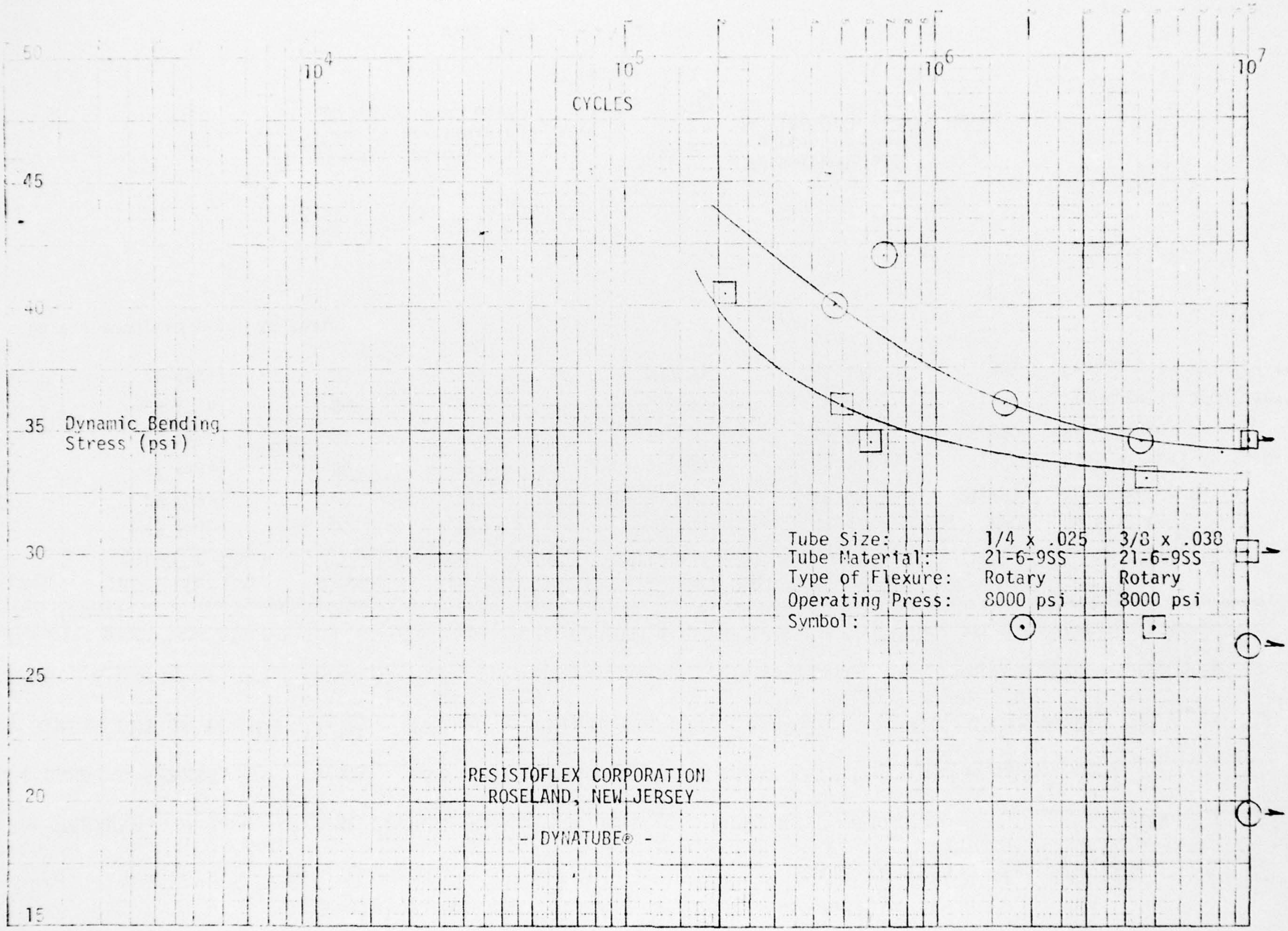

CYCLES
10^4
10^5
10^6
10^7
50
45
40
35
30
25
20
15
Dynamic Bending Stress (psi)
Tube Size: 1/4 x .025 3/8 x .038
Tube Material: 21-6-9SS 21-6-9SS
Type of Flexure: Rotary Rotary
Operating Press: 8000 psi 8000 psi
Symbol:
RESISTOFLEX CORPORATION
ROSELAND, NEW JERSEY
- DYNATUBE® -

APPENDIX B

THERMAL ANALYSIS COMPUTER PROGRAM

FOR MODIFIED T-2C HYDRAULIC SYSTEM

HYDROSA

```
C    T-2C   LHS  HYDRAULIC SYSTEM TEMPERATURE ANALYSIS:
C:
         REAL H,K
         DIMENSION  A(20,20),B(20,1),T(20),IA(20,1)
     100 WRITE(6,1)
       1 FORMAT(1X,'H,TA,BTUHE,OC3,OC8,OD3,OD8,BTU3,BTU8')
         READ(9,*,END=1000)H,TA,BTUHE,OC3,OC8,OD3,OD8,BTU3,BTU8
         DO 110 I=1,20
         B(I,1)=0.0
         T(I)=0.0
         DO 110 J=1,20
     110 A(I,J)=0.0
C:
C    CONSTANTS:
C:
         CP=.55
         K=411.5
         AP3=.479
         AP8=.778
         AP=2.125
         AS3=5.9
         AS8=1.9
         ARET=2.1
         ASUCT3=1.32
         ASUCT8=1.11
         ASUCT=.71
         P3=3000.
         P8=8000.
         AC3=.419
         AC8=.522
         AC =.75
         AR3=6.
         AR8=1.6
C:
C    COEFFICIENTS:
C:
         BTUE=1.486*OD3*P3
         BTUA=1.486*OD8*P8
         B(1,1)=BTU8+H*AP8*TA
         B(2,1)=BTU3+H*AP3*TA
         B(3,1)=H*AP*TA
         B(4,1)=H*AS3*TA
         B(5,1)=+H*AS8*TA
         B(6,1)=H*ARET*TA
         B(7,1)=H*AC3*TA
         B(8,1)=H*AC8*TA
         B(9,1)=H*AC*TA-BTUHE
         B(10,1)=H*ASUCT8*TA
         B(11,1)=H*ASUCT3*TA
         B(12,1)=H*ASUCT*TA
         B(13,1)=0.0
         B(14,1)=0.0
         B(15,1)=16.
         B(16,1)=9.
         B(17,1)=BTUE
         B(18,1)=H*AR3*TA
         B(19,1)=BTUA
         B(20,1)=H*AR8*TA
         OS8=OD8+OC8
         OS3=OD3+OC3
         OC=OC3+OC8
         OP=OD3+OD8+OC
         OS=OP
         A(4,1)=.5*H*AS3-K*OD3*CP
         A(18,2)=.5*H*AR3+K*OD3*CP
         A(14,2)=OD3
         A(5,3)=.5*AS8-K*OD8*CP
         A(20,4)=.5*AR8+K*OD8*CP
         A(14,4)=OD8
         A(6,5)=.5*H*ARET-K*OP*CP
         A(14,5)=-OP
         A(3,6)=H*AP/2.-K*OP*CP
         A(6,6)=.5*H*ARET+K*OP*CP
```

```
      A(7,7)=.5*H*AC3-K*OC3*CP
      A(7,8)=.5*H*AC3+K*OC3*CP
      A(13,8)=OC3
      A(8,9)=.5*H*AC8-K*OC8*CP
      A(8,10)=.5*H*AC8+K*OC8*CP
      A(13,10)=OC8
      A(9,11)=.5*H*AC-K*OC*CP
      A(13,11)=-OC
      A(3,12)=K*OS*CP+H*AR/2.0
      A(12,12)=.5*H*ASUCT-K*OS*CP
      A(10,13)=.5*H*ASUCT8-K*OS8*CP
      A(11,13)=.5*H*ASUCT3-K*OS3*CP
      A(12,13)=.5*H*ASUCT+K*OS*CP
      A(10,14)=.5*H*ASUCT8+K*OS8*CP
      A(11,15)=.5*H*ASUCT3+K*OS3*CP
      A(14,16)=OC
      A(9,16)=.5*H*AC+K*OC*CP
      A(1,3)   =H*AP8/3.0+K*OD8*CP
      A(1,14)  =H*AP8/3.0-K*OS8*CP
      A(2,1)   =H*AP3/3.0+K*OD3*CP
      A(2,15)  =H*AP3/3.0-K*OS3*CP
      A(2,7)=H*AP3/3.0+K*OC3*CP
      A(1,9)=H*AP8/3.0+K*OC8*CP
      A(15,3)=1.0
      A(15,14)=-1.0
      A(16,1)=1.0
      A(16,15)=-1.0
      A(5,19)=H*AS8/2.+K*OD8*CP
      A(4,17)=H*AS3/2.+K*OD3*CP
      A(17,17)=-K*OD3*CP
      A(17,18)=+K*OD3*CP
      A(18,18)=H*AR3/2.-K*OD3*CP
      A(19,19)=-K*OD8*CP
      A(19,20)=+K*OD8*CP
      A(20,20)=H*AR8/2.-K*OD8*CP
      IF(A(3,6).GT.0.0) A(3,6)=0.0
      IF(A(4,1).GT.0.0) A(4,1)=0.0
      IF(A(5,3).GT.0.0) A(5,3)=0.0
      IF(A(6,5).GT.0.0) A(6,5)=0.0
      IF(A(7,7).GT.0.0) A(7,7)=0.0
      IF(A(8,9).GT.0.0) A(8,9)=0.0
      IF(A(9,11).GT.0.0) A(9,11)=0.0
      IF(A(10,13).GT.0.0) A(10,13)=0.0
      IF(A(11,13).GT.0.0) A(11,13)=0.0
      IF(A(12,12).GT.0.0) A(12,12)=0.0
      IF(A(3,16).GT.0.0) A(3,16)=0.0
      IF(A(18,18).GT.0.0) A(18,18)=0.0
      IF(A(20,20).GT.0.0) A(20,20)=0.0
C:
C    SOLUTION (AT=B):
C:
      N=20
      M=1
      F=1.0
      IMAX=20
      KK=ISIMEQ(IMAX,N,M,A,B,F,IA)
      DO 500 I=1,20
  500 T(I)=TT
      BTU3=K*OC3*CP*A(7,1)+K*OD3*CP*A(1,1)+H*AP3*((A(1,1)+A(7,1)+A
(15,1)
     *)/3.-TA)-K*OS3*CP*A(15,1)
      BTU8=K*OC8*CP*A(9,1)+K*OD8*CP*A(3,1)+H*AP8*((A(3,1)+A(9,1)+A
(14,1)
     *)/3.-TA)-K*OS8*CP*A(14,1)
      WRITE(6,6001)
 6001 FORMAT(1X,'OUTPUT DATA T1 THROUGH T20')
      WRITE(6,6002)(A(I,1),I=1,20)
      WRITE(6,6003)
      WRITE(6,6002) BTU3,BTU8
 6003 FORMAT( 1X,'BTU3,BTU8')
 6002 FORMAT(1X,4F7.1)
      GO TO 100
 1000 STOP
      END
```

PARAMETER	CALCULATED SYSTEM TEMPERATURES FOR ORIGINAL T-2C SYSTEM (3000 PSI)																	
	IDLE ON GROUND										STEADY STATE AT 35000 FT							
h, BTU/HR/FT²/°F	1	2	3	4	5	6	7	8			1	2	3	4	5	6	7	8
T_{AMB}, °F	100	←					→	100			0	←					→	0
BTU HE, BTU/HR	0	←					→	0			0	←					→	0
QC3, GPM	.18	←					→	.18			.18	←					→	.18
QC8, GPM	.18	←					→	.18			.18	←					→	.18
QD3, GPM	.1	←					→	.1			.2	←					→	.2
QD8, GPM	.1	←					→	.1			.2	←					→	.2
BTU3, BTU/HR	3000	←					→	3000			4720	←					→	4720
BTU8, BTU/HR	3000	←					→	3000			4720	←					→	4720
	RUN NO. 1										RUN NO. 2							
T1	459	297	240	210	191	178	168	160			558	294	201	154	125	105	91	81
T2	430	267	209	178	159	147	137	130			535	270	177	130	101	82	68	57
T3	460	299	241	211	192	179	169	162			560	295	203	155	126	107	93	82
T4	438	293	245	223	211	204	201	200			542	286	196	150	122	104	90	79
T5	454	280	227	200	185	176	169	165			539	278	187	140	112	93	79	68
T6	385	231	178	153	137	127	120	115			501	241	151	106	79	61	49	40
T7	507	341	288	258	238	225	215	207			648	383	290	242	213	192	178	167
T8	504	341	283	252	232	218	208	199			642	375	281	232	202	181	166	154
T9	509	347	289	259	240	226	216	208			649	384	291	243	214	194	180	169
T10	504	341	283	252	232	218	207	199			641	375	280	231	201	180	164	152
T11	504	341	283	252	232	218	207	199			641	375	281	232	202	181	165	153
T12	458	293	241	211	193	180	170	163			560	296	204	156	128	108	94	84
T13	456	296	239	209	190	177	168	160			558	294	201	154	125	106	92	81
T14	451	290	232	202	183	170	160	153			551	286	194	146	117	98	84	73
T15	450	288	231	201	182	169	159	151			549	285	192	145	116	96	82	72
T16	503	341	282	251	231	217	206	198			640	374	279	230	200	179	163	151
T17																		
T18																		

PARAMETER	CALCULATED SYSTEM TEMPERATURES FOR MODIFIED T-2C SYSTEM (3000 PSI AND 8000 PSI)														
	IDLE ON GROUND										STEADY STATE @ 15000 FT				
h, BTU/HR/FT²/°F	4	—	—	—	4	6	—	—	—	6	5	—	—	—	5
TAMB, °F	120	—	—	—	120	120	—	—	—	120	50	—	—	—	50
BTU HE, BTU/HR	0	3000	6000	9000	12000	0	3000	6000	9000	12000	0	3000	6000	9000	12000
QC3, GPM	.3	—	—	—	.3	.3	—	—	—	.3	.4	—	—	—	.4
QC8, GPM	.64	—	—	—	.64	.64	—	—	—	.64	.74	—	—	—	.74
QP3, GPM	.07	—	—	—	.07	.07	—	—	—	.07	.3	—	—	—	.3
QD8, GPM	.09	—	—	—	.07	.07	—	—	—	.09	.09	—	—	—	.09
BTU3, BTU/HR	3000	—	—	—	3000	3000	—	—	—	3000	4720	—	—	—	4720
BTU8, BTU/HR	8580	—	—	—	8580	8580	—	—	—	8580	11700	—	—	—	11700
	RUN NO. 3					RUN NO. 4					RUN NO. 5				
T1	314	269	224	180	135	261	299	198	166	134	228	199	170	141	111
T2	127	126	125	124	123	128	128	128	128	128	136	124	112	100	88
T3	328	281	235	188	142	276	242	208	175	141	237	208	178	149	119
T4	323	293	263	232	202	283	264	246	227	209	216	198	181	163	145
T5	335	281	230	178	127	284	244	205	165	126	240	207	174	141	109
T6	326	276	226	176	127	276	238	201	163	126	234	202	170	139	107
T7	341	298	254	211	167	288	258	227	197	166	260	231	203	174	146
T8	336	293	251	207	166	282	253	223	194	164	255	227	199	172	144
T9	364	319	273	228	182	311	279	246	214	181	285	256	227	198	169
T10	360	316	271	226	182	307	275	243	212	180	281	253	224	196	167
T11	353	309	265	221	177	299	268	237	206	175	272	244	215	187	159
T12	319	271	223	174	126	268	232	197	161	125	228	198	167	136	105
T13	317	269	221	174	126	265	230	195	160	125	227	196	166	135	105
T14	312	265	219	172	126	257	226	192	159	125	221	192	162	133	104
T15	305	260	215	171	126	252	220	189	157	125	219	190	161	132	102
T16	349	292	235	177	120	296	251	207	162	118	269	229	190	151	111
T17	148	142	135	129	122	127	127	127	127	127	164	146	127	108	89
T18	168	162	155	148	142	146	146	146	146	146	184	166	147	128	109
T19	301	264	226	189	152	258	233	208	183	157	202	180	157	134	111
T20	353	316	278	242	204	310	285	260	235	210	255	232	209	186	164